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Service Experience of MAN B&W Low Speed Diesel Engines

Introduction

A very large number of MC & ME engines are entering service these years. The latest development of the most successful marine engine series ever is the ME-B series of which more than hundred engines are on order or delivered. The ME-B series are targeting the small bore end (35, 40, 46, 50 and 60) of the MAN B&W low speed engine range.

Electronically controlled low speed diesels have been part of our engine programme for several years, actually since 2001. Today, approx. 500 electronically controlled engines are in service, and focus on the electronically controlled engine versions is expected to increase with IMO Tier II emission rules coming into force for vessels with keel-laying after 1 January 2011. Also, because of this development the optimised ME-B engine range is very important and has already grabbed a lot of attention among shipowners.

This paper deals with the latest service experience obtained so far from ME/ME-C engines in service. Early service experience for the 6S40ME-B will also be discussed.

Advantages of upcoming ME software updates focusing on onboard trouble shooting will be described and related to service experience.

Furthermore, an update on service experience of the MC/MC-C engine series will be given, focusing on the engine structure.

Common for both the ME/ME-C and the MC/MC-C engine series is the well-documented possibility to do Condition Based Overhaul (CBO) with an average Time Between Overhauls (TBOs) of 32,000 hours and higher. For tankers, this opens up the possibility of doing only major overhauls at dockings with 5-year intervals. Many shipowners have now had the positive experience of CBO. Also the development in relation to cylinder condition with focus on cylinder oil consumption will be touched upon.

Due to the present economic crisis, focus has lately generally been devoted to optimisation of low load operation. In early 2009, MAN Diesel & Turbo issued a Service Letter dealing with the possibility of operating continuously down to 10% load. Service tests with various scavange air pressure increasing measures at low load have also been carried out. Here, tests with turbocharger cut-out, Variable Turbine Area (VTA) turbochargers and exhaust gas bypass are the most important ones. The test results will also be discussed.

**Service Experience Update
ME System**

Over the past years, we have described various areas with room for improvement on the ME system in a number of papers dealing with service experience. The most recent paper is Service Experience 2008, MAN B&W Engines, Ref. [1].

In this paper, we will comment on issues which are still undergoing investigation at the time of writing. Also, future planned upgrades will be mentioned. Especially the trouble-shooting tools implemented in a new ECS software version will be commented on.

Cavitation damage in the exhaust valve actuation system

Fig. 1 shows cavitation damage in the high pressure pipe between the exhaust actuator and the exhaust valve. Also cavitation damage in the actuator top cover and in the top of the exhaust valve can be seen. This kind of damage is seen on the large bore versions of the ME engine (80, 90 and 98-cm bore),

however, not to the same extent on all units on the engine. Therefore, damage can be counteracted by relatively small changes.

Service tests where the proportional feature of the FIVA valve is used are

presently ongoing. So far, we have only used the proportional control of the FIVA valve for injection rate shaping, but now we also use it in order to open and close the exhaust valve gently. Fig. 2 shows the original lifting curve and the modified lifting curve of the exhaust valve. No

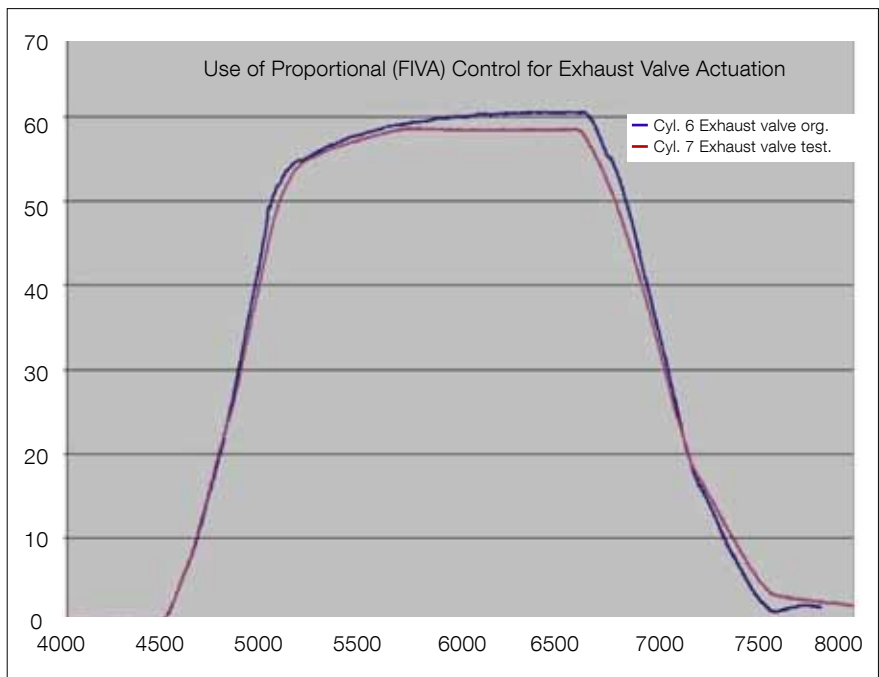


Fig. 2: Use of proportional (FIVA) control for exhaust valve actuation

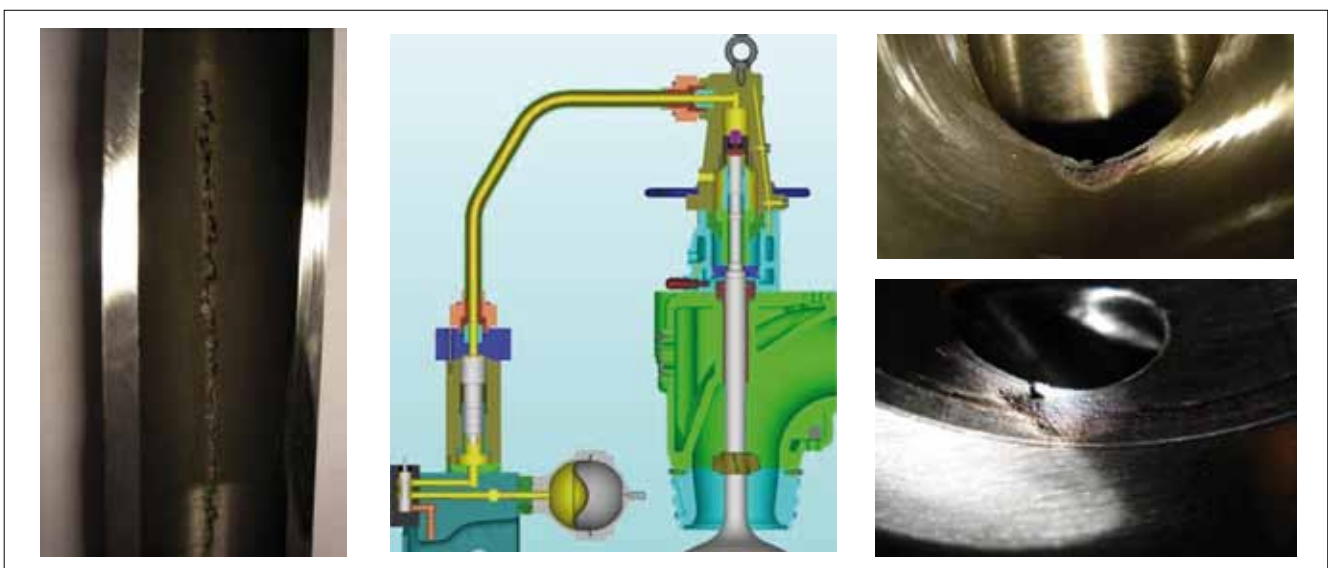


Fig. 1: Cavitation in hydraulic exhaust actuation system

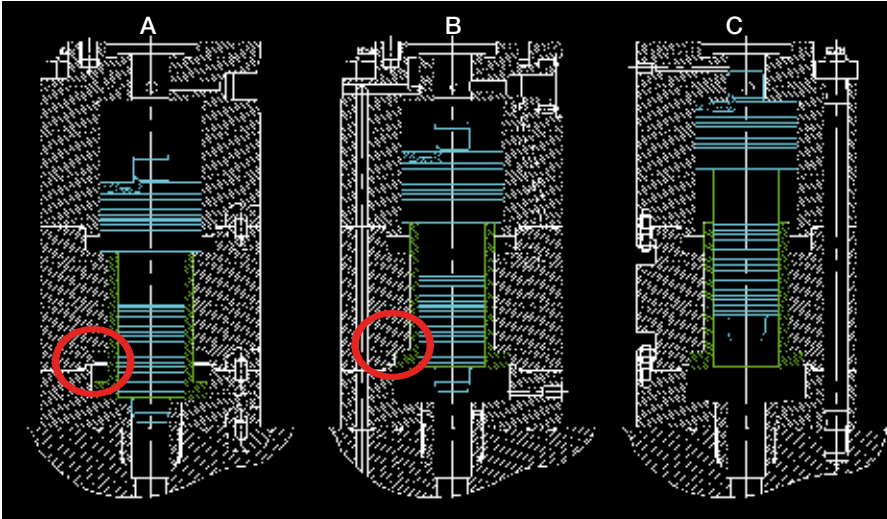


Fig. 3: Exhaust actuator

big changes can be seen. However, the oil pressure fluctuations in the actuator system are modified to some extent, and we expect that this will reduce/eliminate the cavitation damage in the exhaust actuation system.

Exhaust actuator piston breakage

The exhaust actuator for the ME engines is a two-step actuator. The principle of function is shown in Fig. 3. Fig. 3 (A) shows both pistons (step 1 and step 2) in bottom position. Fig. 3 (B) illustrates

how the step 1 piston is stopped in the “damper” as the step 2 piston continues its movement upward. In Fig. 3 (C), the step 2 piston has arrived at its top position caught by another “damper”.

On a few actuators for the original 200 bar hydraulic system, we have seen breakage of the step 1 piston collar, Fig. 4. We are convinced that this problem relates to delayed “take-off” of the step 1 piston from the bottom position, Fig. 3 (A). This is caused by a “glue” effect, which exists from time to time. When the step 1 piston is delayed it will create very high pressures in the “damper” chamber and hereby increased load on the piston collar is generated. Various means to avoid the “glue” effect can be introduced by generating a force by the oil pressure under the step 1 piston. An example of retrofit is illustrated in Fig. 5. Such design modifications will eliminate the trouble with step 1 piston breakage.

Cavitation in the pilot step of the FIVA valve

We have seen cavitation attacks on the pilot spools of both the Curtis Wright



Fig. 4: Piston collar breakage

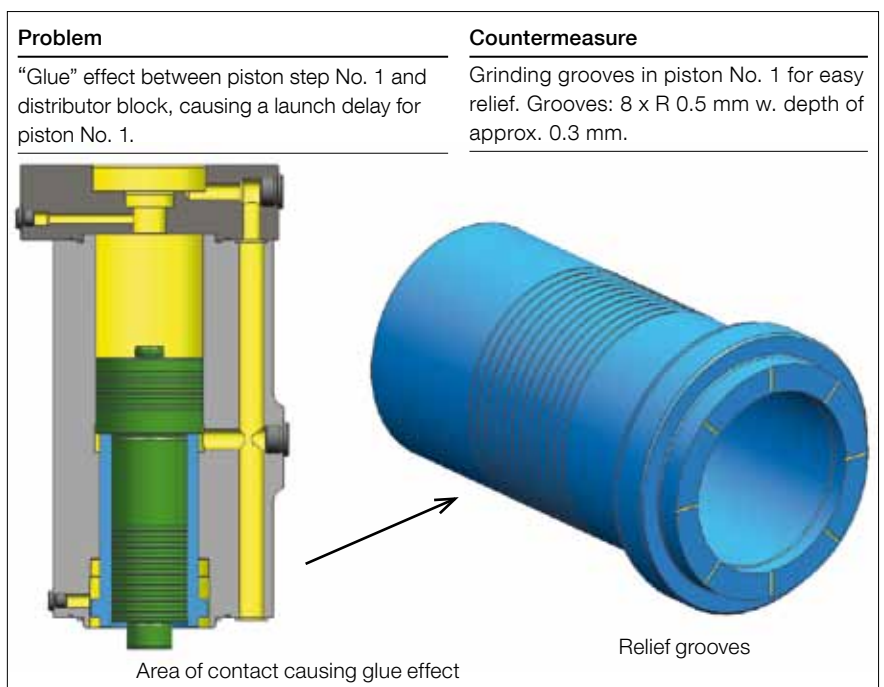


Fig. 5: Example of retrofit

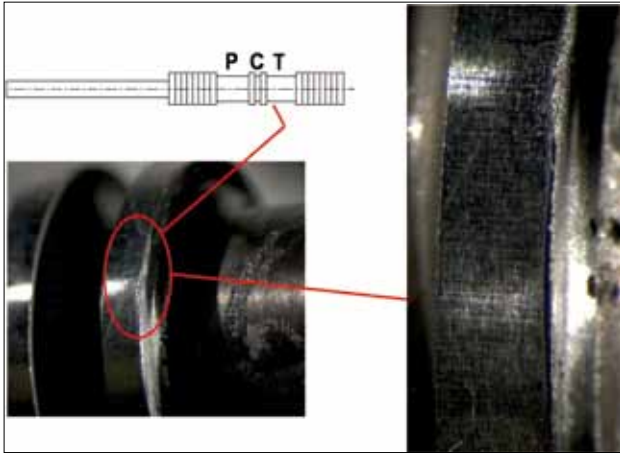


Fig. 6: Curtis Wright FIVA: cavitation on pilot spool

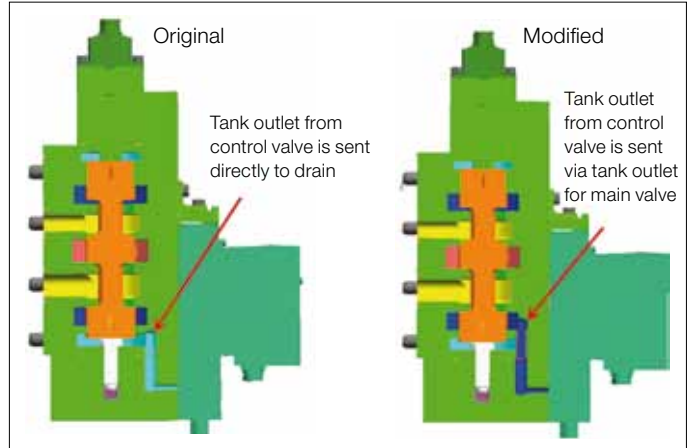


Fig. 8: MAN B&W FIVA: Cavitation countermeasure test 1

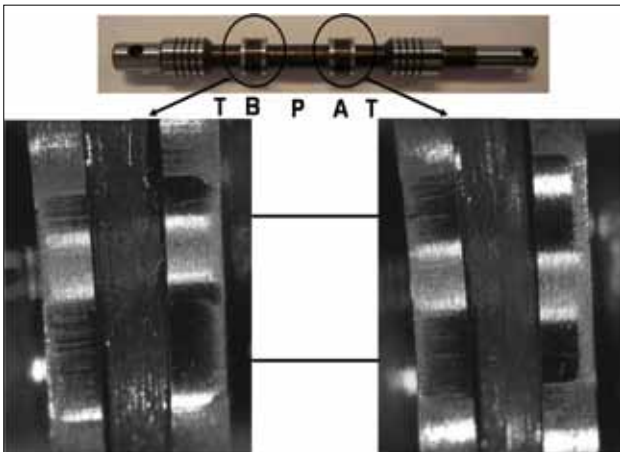


Fig. 7: MAN B&W FIVA: cavitation on Parker valve pilot spool

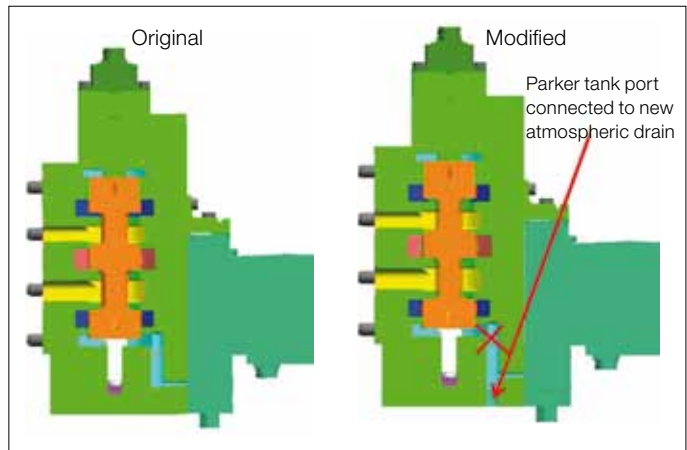


Fig. 9: MAN B&W FIVA: Cavitation countermeasure test 2

FIVA valve, see Fig. 6, and the Parker valve on the MAN B&W FIVA valve, see Fig. 7. Presently, this limits the overhaul intervals of the FIVA valves to some 8,000-10,000 hours. Our goal for overhaul intervals on FIVA valves is 32,000 hours.

The cavitation damage on the Parker spools is mainly related to the tank ports for the pilot step. We have therefore designed two (2) service tests, which are currently accumulating hours on two (2) 12K98ME plants. Fig. 8 shows the test

FIVA with the pilot tank port connected to the FIVA main spool pressurised tank port. Fig. 9 shows the other FIVA valve with the pilot tank port connected to a completely pressure-less drain.

On the basis of these test results, we will conclude final countermeasures in order to extend overhaul intervals to an acceptable time span. Preliminary inspection results from the service test indicate that cavitation countermeasure test 1, the pressurised submerged drain, is the most successful solution.

However, more service hours are needed to make the final conclusion.

Untimed injection with the MAN B&W FIVA valve

For the MAN B&W FIVA valve, we have experienced an untimed injection in one single case on one cylinder unit on a 6S60ME-C engine in August 2009. The over-pressure in the cylinder resulting from the untimed injection did not cause any damage to the mechanical parts because of the built-in safety margin in the various components, such

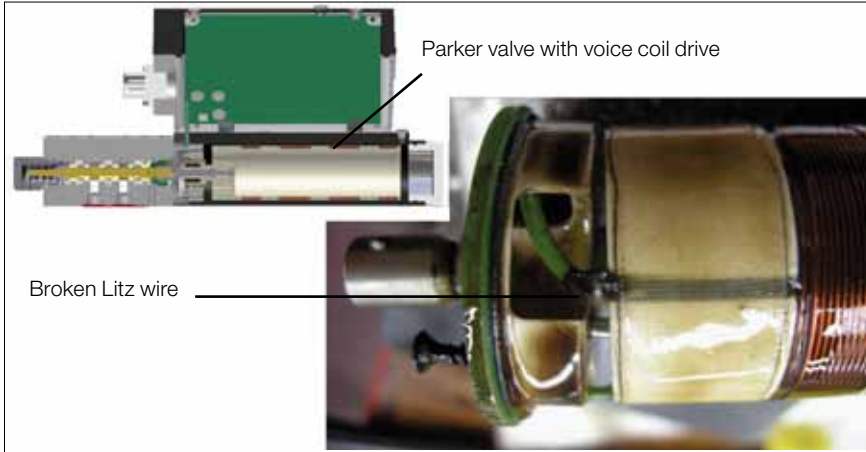


Fig. 10: Parker pilot valve

as the piston rod, connecting rod and crankshaft. The failing FIVA valve was returned, and investigations showed that the problem was related to the pilot step, the Parker valve. Close investigations together with the sub-supplier revealed that a broken wire, a so-called Litz wire, had caused the malfunction of the Parker valve, see Fig. 10.

Tests also revealed that the broken Litz wire resulted in an overshoot and that explained the untimed fuel injection experienced in the vessel with the 6S60ME-C engine.

Production of the Parker valves was changed based on this experience, and zener diodes were included in the electronic control design of the Parker valve. These diodes eliminate overshoot in case of a broken Litz wire.

Investigations of Parker valves with different numbers of operating hours have been concluded, and Parker valves with low serial numbers have been recalled from service.

Multi Purpose Controller (MPC) quality

In order to improve the quality of the MPCs, we have introduced so-called “burn-in” tests of all MPCs produced. The background is an investigation of more than 100 returned MPCs. Of these, approx. 25% did not show failure until they were subjected to the “burn-in” test. With the introduction of full scale burn-in tests, we expect that the MPC failure rate will be reduced sig-



Fig. 11: New ECS software with trouble shooting screens: HCU event recorder

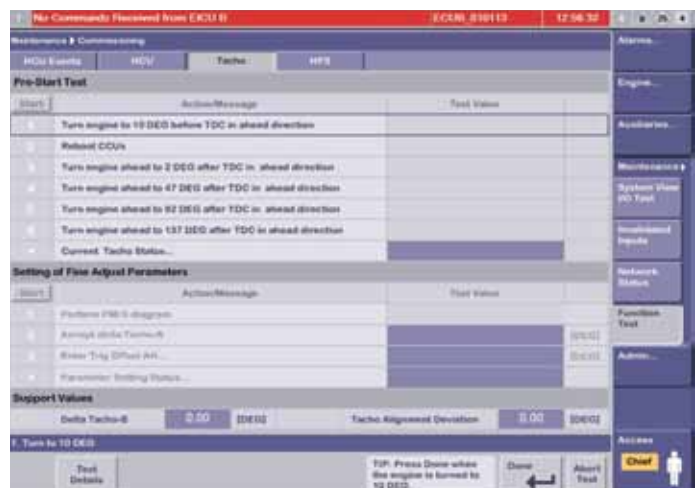


Fig. 12: New ECS software with trouble shooting screens: Tacho adjustment

nificantly, especially during the commissioning stage (shop test, quay trial and sea trial).

New updated ECS software

A new version of the Engine Control System (ECS) software will soon be introduced both to new engines and to engines in service. The main focus for this software is to provide better trouble-shooting tools onboard vessels equipped with ME engines.

Various new screens on the MOP have been developed in order to assist the engine crew in more qualified trouble shooting. An example is the “HCU event recorder” and the related MOP screen, see Fig. 11. The “HCU event recorder” records a number of predetermined signals related to the HCU (Hydraulic Cylinder Unit) continuously. If an alarm

related to the HCU is activated, a record of signals is stored and can later be seen on the MOP, some r/min before activation of the alarm and some r/min after. This will assist the engine crew in locating the reason for the alarm.

Another example of improved support to the engine crew is the tacho adjustment screen, see Fig. 12. This MOP screen assists the engine crew in making re-adjustments to the tacho system.

There are a number of other new screens relating to the HCU and the HPS (Hydraulic Power Supply). Altogether, the new software will enable much more qualified trouble shooting onboard.

Off-engine located MPCs

As an alternative to the MPCs located on the engine, we have introduced a design where the MPCs are located in larger cabinets, see Fig. 13, which can be located away from the engine, e.g. in the engine control room, in a switch-board room or directly in the engine room. From now on, we will gain experience with respect to i.a. lifetime of the MPCs. Also, possible production benefits of the alternative execution can be measured in the future.

Drain box for the exhaust valve drain lines

On 80-cm and 98-cm bore engines, we have seen breakage of the drain lines from the exhaust valve. A design with a so-called drain box, see Fig. 14, has been tested successfully. This design will counteract the trouble with drain line breakage.



Fig. 13: Large cabinet with MPCs suited for engine

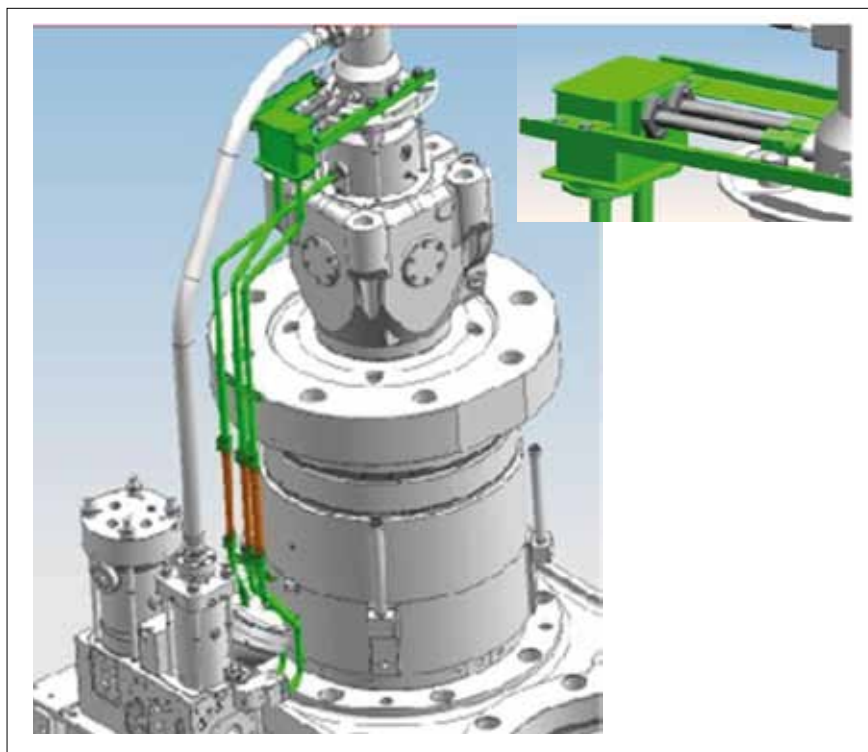


Fig. 14: Exhaust valve drain line with drain box

First Service Results for Mark 9 Large Bore and ME-B Engines

7K80ME-C9 Service Experience

At the time of writing, the first four large bore 7K80ME-C9 engines are in service.

Combustion chamber components have been updated with additional features for the high-rated large bore Mark 9 engine series. Some of these features are:

- Pitch-honed cylinder liners
- CPR top piston ring with a thick alu-coat on top of cermet coating
- Mo-coated barrel-shaped piston skirt
- Piston cleaning ring of asymmetric design.

Pitch honing means that the semi-honing has been updated to include honing of the liner surface grooves (wave cut).

A thick alu-coat gives more safe running in properties of the piston ring package.

Mo-coated barrel-shaped piston skirts have been tested with good results for a long time on a number of large bore engines. Cost down potential exists for this design.

An asymmetric piston cleaning ring gives a better cleaning effect on the piston top land in the exhaust side.

Scavenge port inspection has been carried out after 1,842 running hours. The results can be seen in Figs. 15 and 16 for piston ring and piston skirt condition, respectively. Fig. 17 shows the pitch-honed liner from the scavenge ports. All inspection areas indicate that



Fig. 15: Piston ring condition



Fig. 16: Piston skirt view

very successful running-in has taken place and that the cylinder condition for all units is excellent.

A further inspection was carried out after 3,800 running hours. One cylinder unit was opened up for inspection and liner calibration. Fig. 18 shows the top of the liner. The max. wear rate is 0.037 mm/1000 hrs. at the top dead centre of piston ring No. 2. This wear rate includes initial running-in wear and is very satisfactory.

Fig. 19 shows a close-up of the cylinder liner surface. The open graphite structure is clearly seen. This surface structure indicates a successful temperature control on the liner surface and, hereby, a controlled acid corrosion level.

A new main bearing design concept has been introduced on the 7K80ME-C9. The result of this new design concept is improved productivity during machining of the bedplate. Figs. 20 to 23 show the results of the overhaul inspection of main bearing No. 5 after 3,800 hours.



Fig. 17: Pitch-honed liner



Fig. 18: Top of liner after 3,800 hrs.

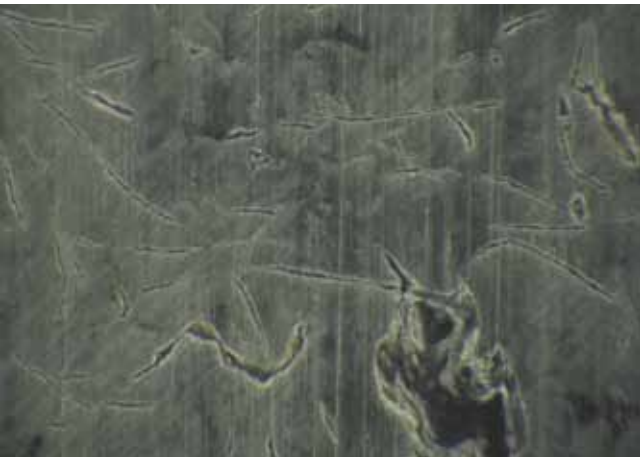


Fig. 19: Close-up of cylinder liner: open graphite structure visible



Fig. 20: High-friction shim and guide pin



Fig. 21: Main bearing cap and bedplate contact area



Fig. 22: Upper main bearing shell backside



Fig. 23: Lower and upper shell, white-metal side



The inspection results indicate excellent condition with respect to contact surfaces, with signs of relative movements as well as very good main bearing shell load imprints.

ME-B9 service experience

At the time of writing, two 6S40ME-B9 and two 8S35ME-B9 engines are in service. A scavenge port inspection was made after 2,574 hours of operation on the 6S40ME-B9 prototype. Fig. 24 shows pictures from this inspection.

The pistons are equipped with an aluminum ring pack. As shown, running-in has finished, and the piston ring base material is working directly against the cylinder liners. Light piston top-land deposits are seen.

We are using the S40ME-B9 engine prototype for cylinder oil feed rate tests. The aim is to come down to the feed rate used as standard for >60 bore engines (0.20 g/kWhXS%, minimum 0.6 g/kWh). These tests look very promis-

ing, and a new service letter for small bore engines on reduced cylinder oil feed rate is expected soon.

A piston overhaul was carried out after 4,000 running hours to inspect the condition of the new piston design, see Fig. 25, applied on the S40ME-B9 engines. Fig. 26 indicates a very good condition of the piston top-land (light deposits), piston rings (surface without any signs of seizures) and piston skirt (perfect condition of the bronze band).

Unit No. 1:

Running hours:	2,574
Condition:	Good. No signs of abnormal wear or micro-seizures. No alu-coating left
Present cylinder oil dosage:	0.75 g/kWh
Recommended cylinder oil dosage:	Keep lubricating according to the cylinder oil test by following service letter SL09-507
CL groove:	Not accessible
Topland:	Light deposits



Fig. 24: Scavenge port inspection



Fig. 26: ME-B9 piston after 4,000 hrs

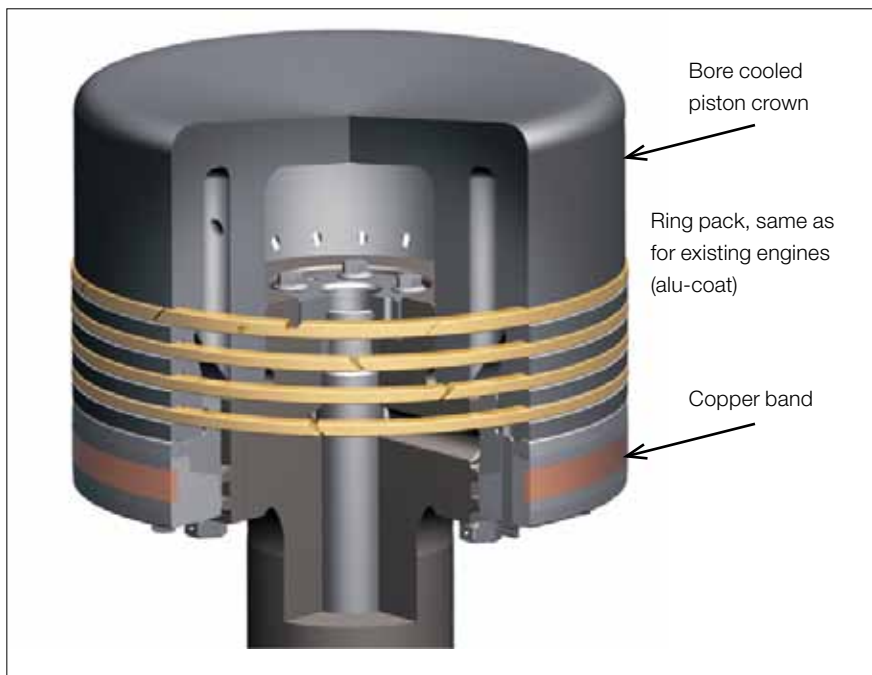
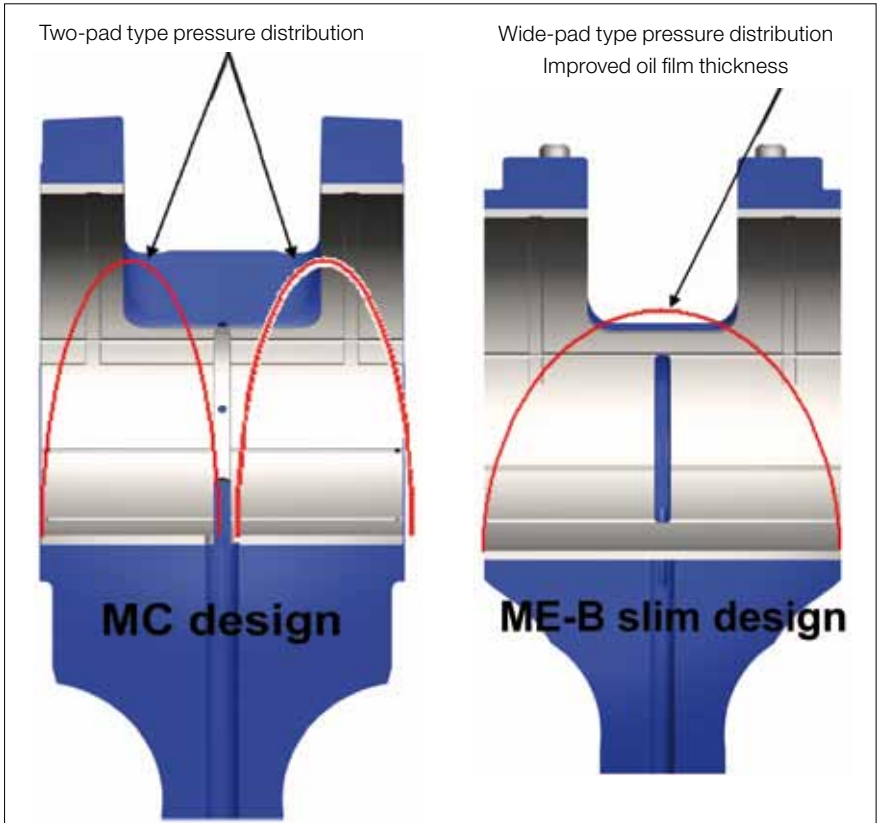


Fig. 25: ME-B9 piston design



A new design of the slim type connecting rod and the wide-pad crosshead lower bearing shell, see Fig. 27, has been introduced on the ME-B9 engines. Fig. 28 shows the lower crosshead bearing shell of unit No. 6 at an inspection after 4,000 running hours. The bearing imprint is, as expected, in good condition.

Fig. 27: ME-B9 crosshead bearing design



Fig. 28: ME-B9 crosshead bearing after 4,000 hrs.

General Update on ME and MC Engines

Apart from ME system specific issues, we also have certain areas of attention, which are common for the electronically controlled ME engines and the cam-shaft controlled MC engines. In this paper, we also focus on the cylinder condition of large bore engines, lifting bracket cracks on large bore engine bedplates and cracks in the first generation of welded cylinder frames.

Cylinder condition, large bore engines

In general, we experience very satisfactory wear figures on both cylinder liners and piston rings on large bore engines. This has enabled an extension of overhaul intervals, and strategy overhaul intervals, above 32,000 hours (5 years) can be obtained when applying condition based overhaul (CBO). This is described in our service letter SL07-483/HRR, Ref. [2], and further in the paper Service Experience 2008, MAN B&W Engines, Ref. [1]. We will deal with condition based overhaul strategy in details in the next section of this paper. However, from time to time, the generally good wear figures are disturbed by cylinder liner scuffing. The reasons for cylinder liner scuffing are many, and often caused by several factors. The major reasons for cylinder liner scuffing include:

- Bore polish due to surplus of alkali additive (excessive lubrication)
- Broken down oil film as a result of too rapid load changes
- Water ingress due to inefficient water separation

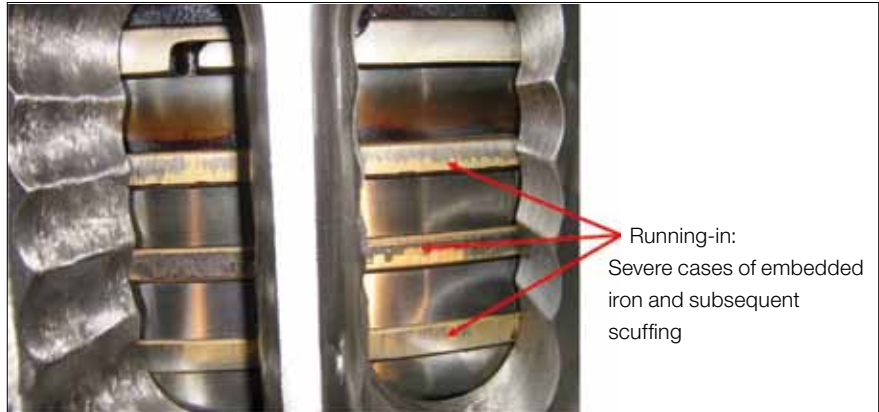


Fig. 29: Increased focus on piston ring quality and liner surface quality

- Catfines in the fuel – wear out of the CL grooves/broken rings
- Running-in problems.

Lately, we have had to focus on running-in problems, typically at running hours between 500 and 1,000. The reason for the running-in problems is the piston ring quality in combination with the cylinder liner surface quality. Fig. 29 shows a severe case of running-in problems due to embedded iron on

the outside of the running-in alu-coat layer, peeling-off of the alu-coat/iron layer and subsequent scuffing. A detailed analysis of such problems has led to increased focus on:

- Piston ring quality
- Liner surface quality/finish.

Fig. 30 shows cross sections of piston rings with iron and cermet (delaminated) layers positioned on top of the alu-coat

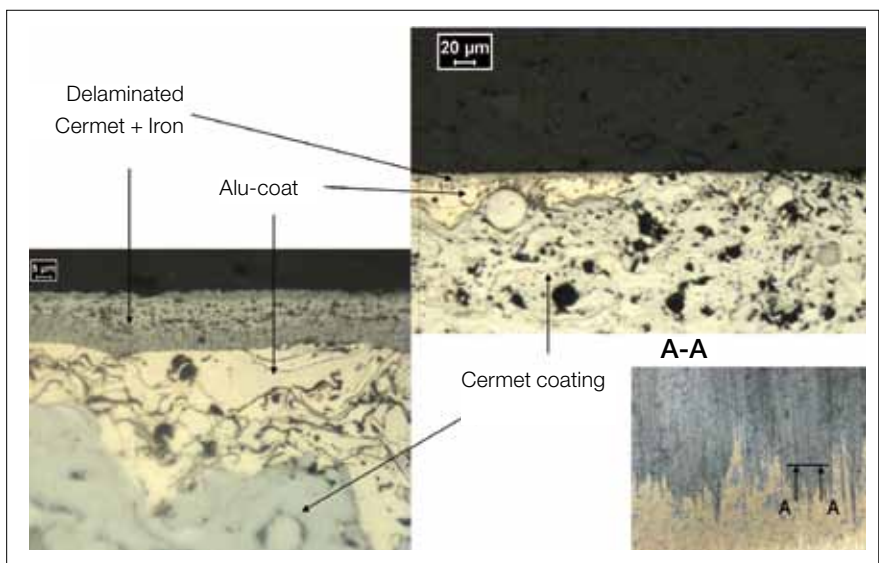


Fig. 30: Piston ring quality

layer during running-in, leading to unstable cylinder condition caused by the quality of the cermet coating.

Fig. 31 shows an example of off-spec. honing of the cylinder liner surface. The honed area must nominally be 50% of the liner surface. As can be seen on the photo, the honed area is much smaller resulting in much tougher running-in and an increased production of 'liner-iron'. The photo shows an unused spare cylinder liner on a vessel where cylinder liner scuffing has occurred.



Fig. 31: Cylinder liner quality: insufficient plateau honing

During the last part of 2009 and the first part of 2010, we carried out a piston ring exchange programme where around 1,000 cylinder units of K98 and K90 engines were exchanged before ship delivery in order to gain control in relation to piston ring quality. The cermet coating specification and the quality of the cermet coating were reformulated. During this process also cylinder

liner machining was checked and re-honing was carried out in case of off-spec. condition.

Another recent focus area has been the design and efficiency of water mist catchers. Fig. 32 shows the full scale

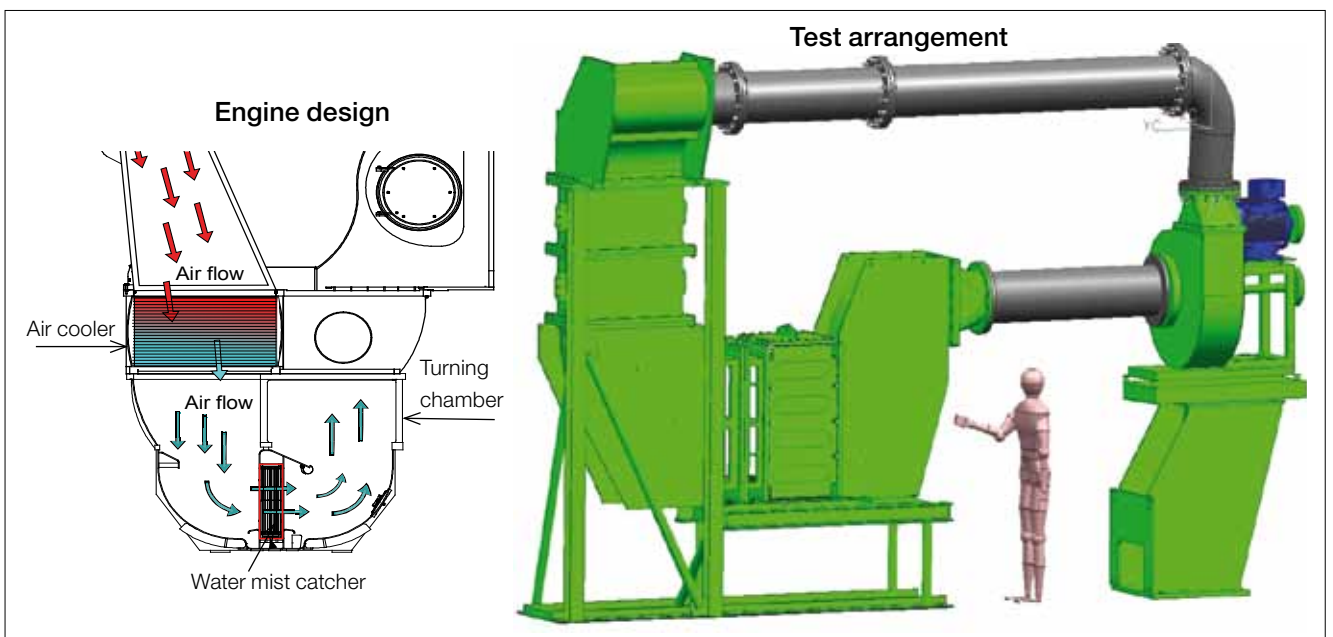


Fig. 32: Full scale test installation

test installation of a water mist catcher. We have made a full scale test on various water mist catcher designs, including some sub-supplied designs, which have been applied in big numbers in our large bore engines. Fig. 33 shows three of the tested designs and Fig. 34 shows an example of the results obtained. In

this case, the water mist catching efficiency is shown as a function of air velocity and very big deviations in efficiency can be seen.



Fig. 33: Tested designs

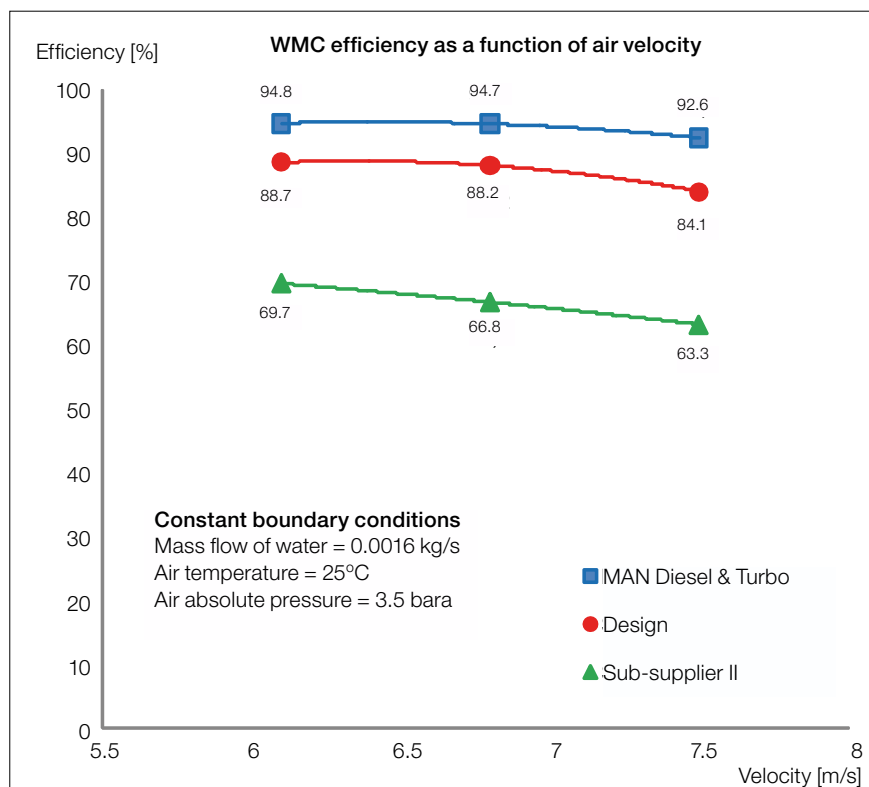


Fig. 34: Full scale test results

On the basis of these results and an update of the mechanical design, we have now fully re-specified the water mist catcher design, and we expect that this will limit water ingress related scuffing.

Cracks in the bedplate lifting bracket of K98 engines

In 2008, cracks in the lifting bracket on K98 bedplates were discovered on engines produced 5-7 years earlier, see Fig. 35. Soon after the first cracks were discovered, a so-called 'Circular Letter' was issued to all owners/operators of K98 engines, see Fig. 36. In this letter we asked for help to inspect for lifting bracket cracks. Furthermore, we informed about preventive countermeasures, which were burr grinding of the weld seams on the aftmost and foremost brackets. Also, a repair procedure was developed when the cracks first occurred. A modified bracket profile has been designed for new engines, see Fig. 37.

Due to the high number of K98 engines in service, these rectifications are still ongoing and, fortunately, we are able to perform this work without disturbing the operation of the vessels involved.



Fig. 35: Lifting bracket cracks

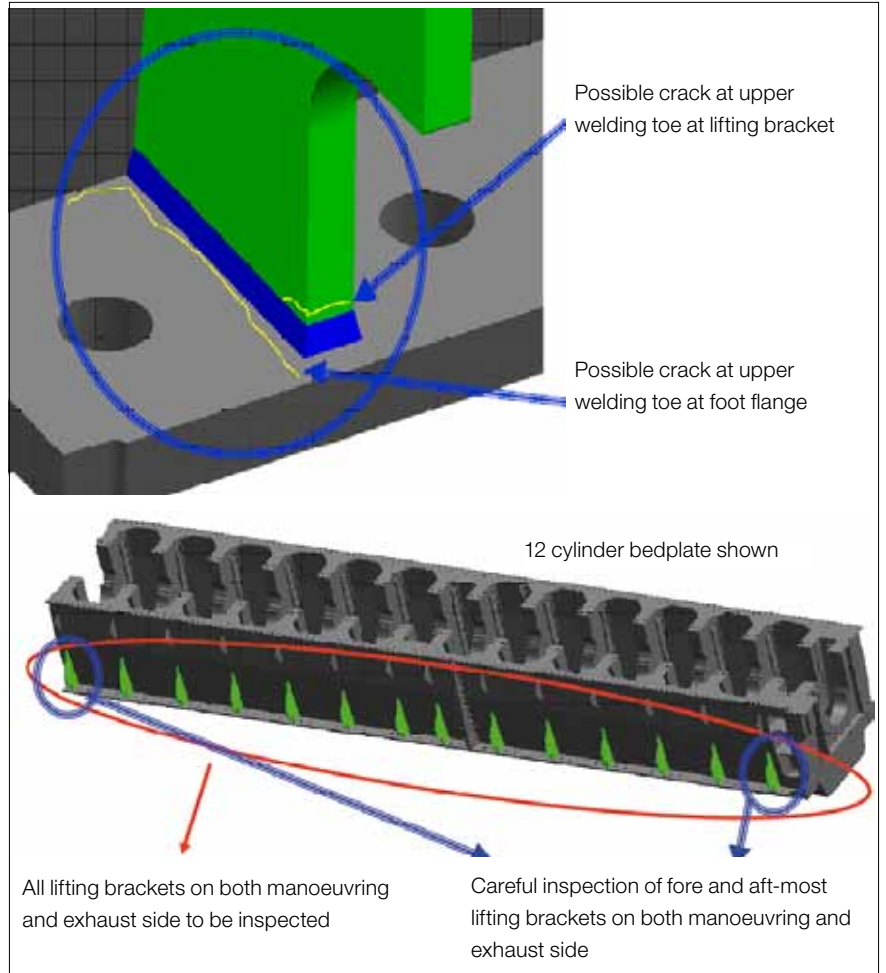


Fig. 36: Circular letter: K98 lifting bracket cracks

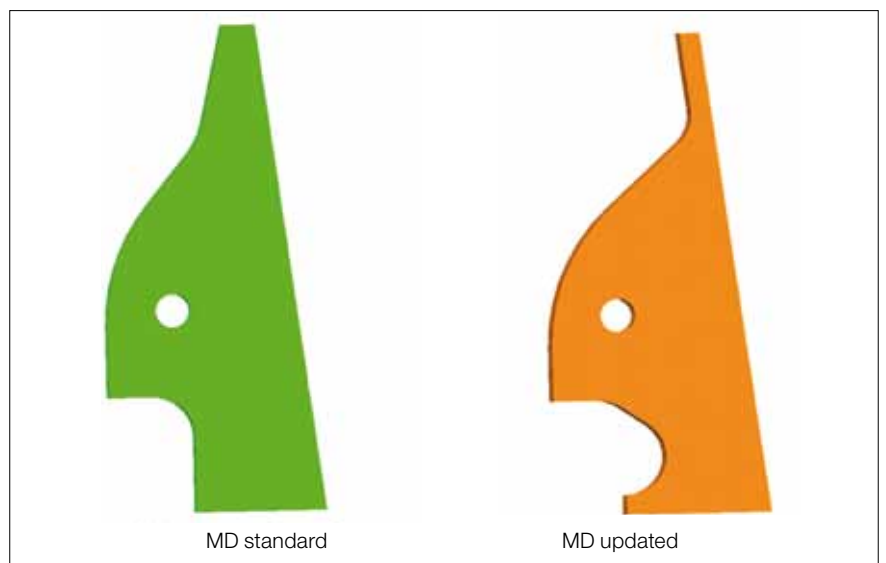


Fig. 37: Lifting bracket cracks: design update

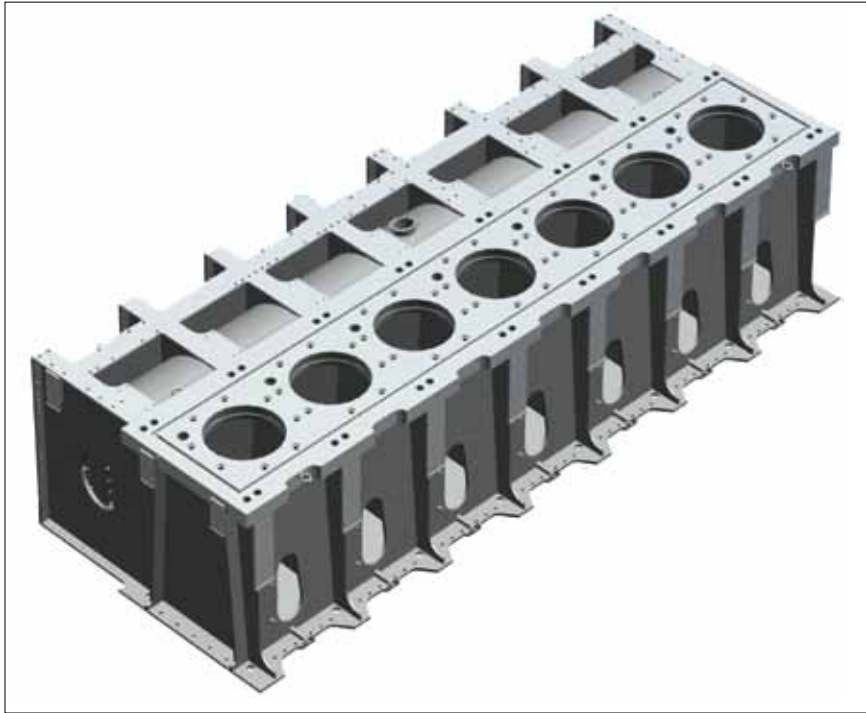


Fig. 38: Welded cylinder frame with integrated scavenge air receiver

Welded Cylinder Frames

As an alternative to cast iron cylinder frames, a welded version of the cylinder frame has been introduced. Fig. 38 shows a 7-cylinder welded cylinder frame with an integrated scavenge air receiver. The first generation of the welded cylinder frame showed, in some cases, cracks originating from the stay bolt covers, see Fig. 39. A new bent type stay bolt cover has been designed, and attachment to the main cylinder frame structure at a position with a lower stress level has been realised.

Fig. 40 shows the newest version of the welded cylinder frame for a 80ME-C Mk 9 engine. On the exhaust side, the bent type stay bolt cover is applied, and on the pump side the stay bolt cover is omitted.

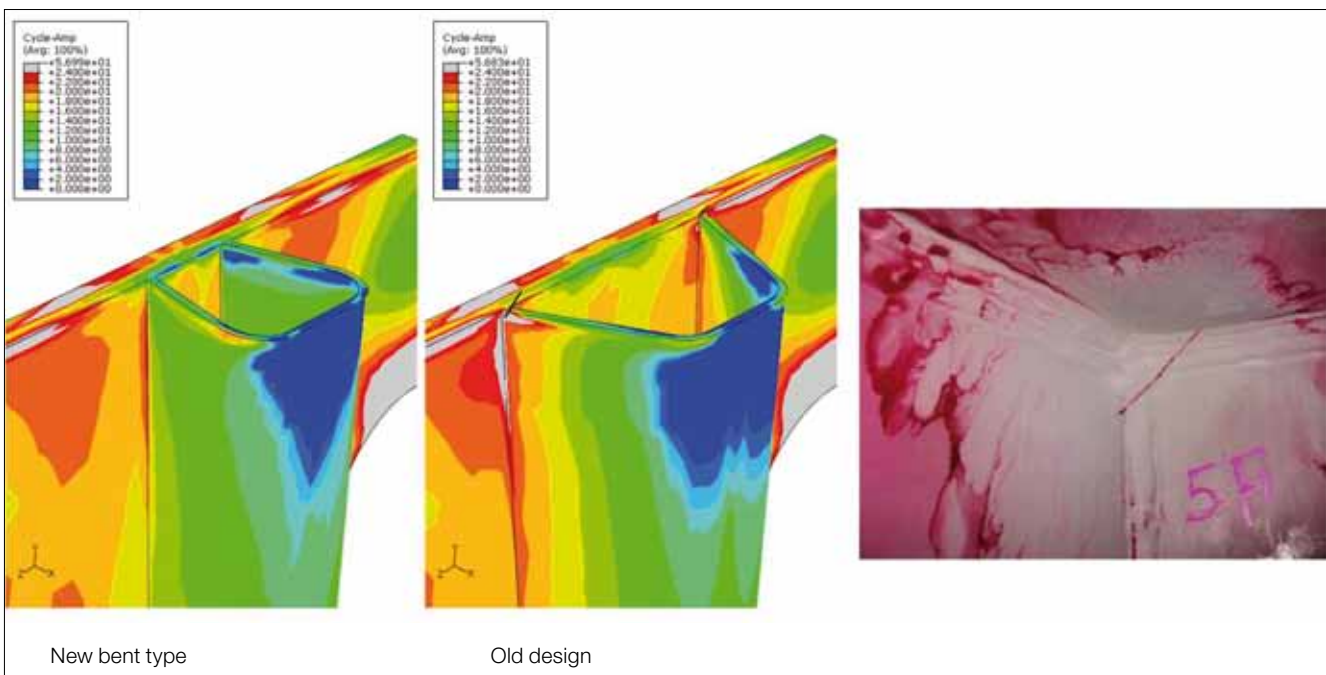
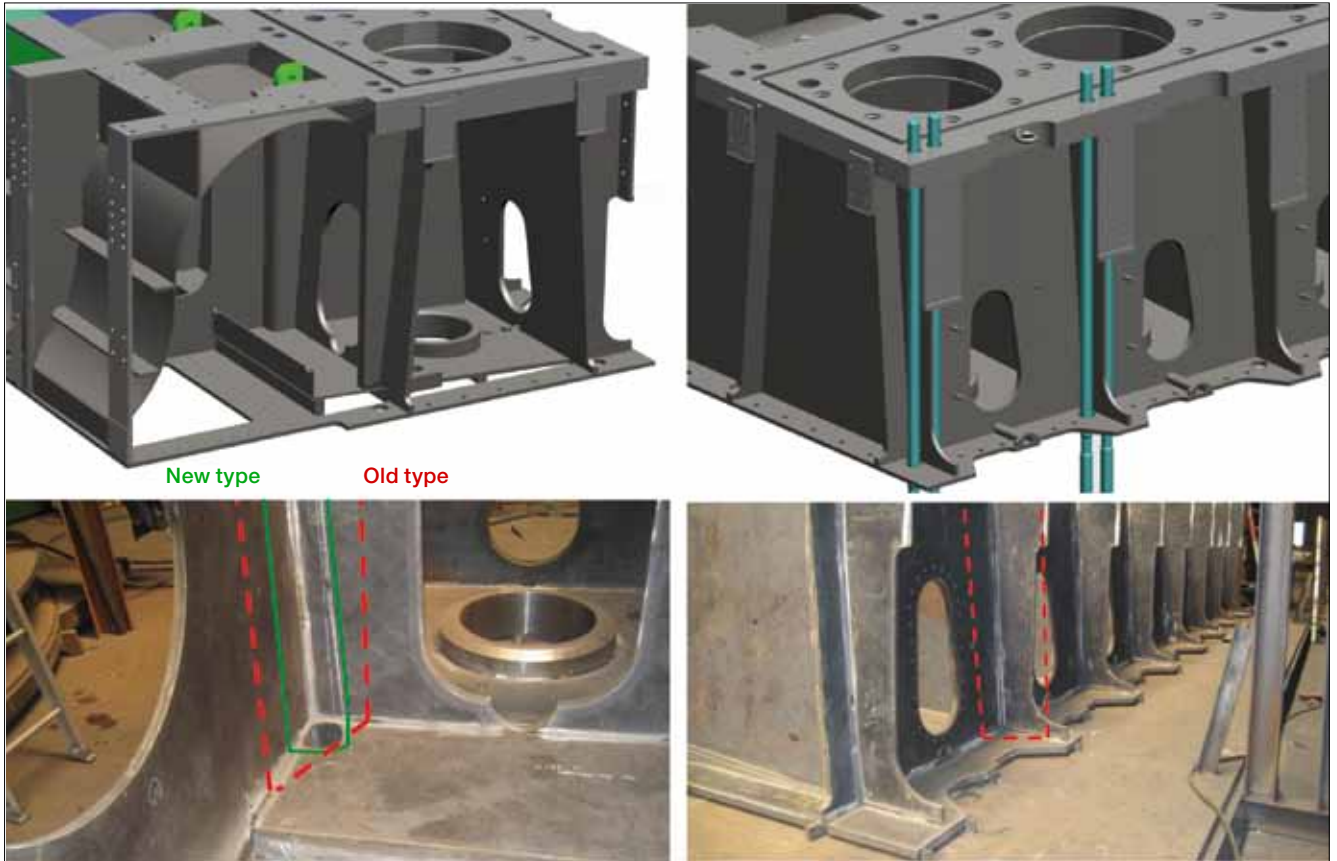


Fig. 39: Welded cylinder frame: fatigue crack originating from stay bolt covers



Thin, bent stay bolt cover in exhaust side. **New type/old type**
 Fig. 40: Welded cylinder frame: old and new type stay bolt cover designs

Pump side stay bolt cover omitted. Stay bolt visible

We are confident that the crack problems related to the stay bolt cover plates have now been eliminated. For engines in service with first generation welded cylinder frames, repair work is in progress.

Condition Based Overhaul Strategy

In the following, we will describe the recent service experience of the MC/MC-C engine series, with focus on condition based overhaul (CBO) and update of monitoring systems. CBO is of course also relevant and possible for ME/ME-C engines.

Condition Based Overhaul (CBO) of Pistons

Experience with our engines with the latest updated combustion chamber design, i.e. with Oros shape and the latest piston ring design, slide fuel valves and optimised temperature levels counts more than eight years of operation. Against this background, we have gained valuable knowledge about the need for piston overhauls compared with earlier experience.

The “Guiding Overhaul Interval” for pistons, previously set to 12-16,000 hours, appears to have been set too conservatively. Normally, the need for piston overhaul does not arise until much later, and extensions up to 32,000 hours are possible. However, the fact is that the scatter is large, and many factors are decisive for the need for overhaul.

This calls for a CBO strategy, the objective being to obtain the highest number possible of safe running hours. Preferably, overhauling should only be carried out when necessary.

The most important factor in a CBO strategy is the evaluation of the actual condition, by means of regular scavenge port inspections and logging of wear and hot corrosion. All the decisive factors for piston overhaul can be

checked via inspections through the scavenge air ports.

The most important factors for piston overhauls are (see Fig. 41):

- Piston ring wear
- Max. amount of hot corrosion allowed on piston top on the centre part (where it is normally highest) is 9/12/15 mm on, respectively, 80/90/98-cm bore engines
- Ring groove clearance. Max. clearance recommended is 1.0 mm on 80 and 90-cm bore engines, and 1.1 mm on 98-cm bore engines

- Sticking, broken or collapsed piston rings or leaking pistons

- Macro-seizures on piston ring running surfaces.

Inspection and logging of the actual cylinder condition and wear should be performed regularly to become familiar with the wear-and-tear development in the cylinder. At the beginning, intervals should be short, e.g. every second to third week. The intervals can be prolonged as confidence builds up.

The following factors should be measured and recorded:

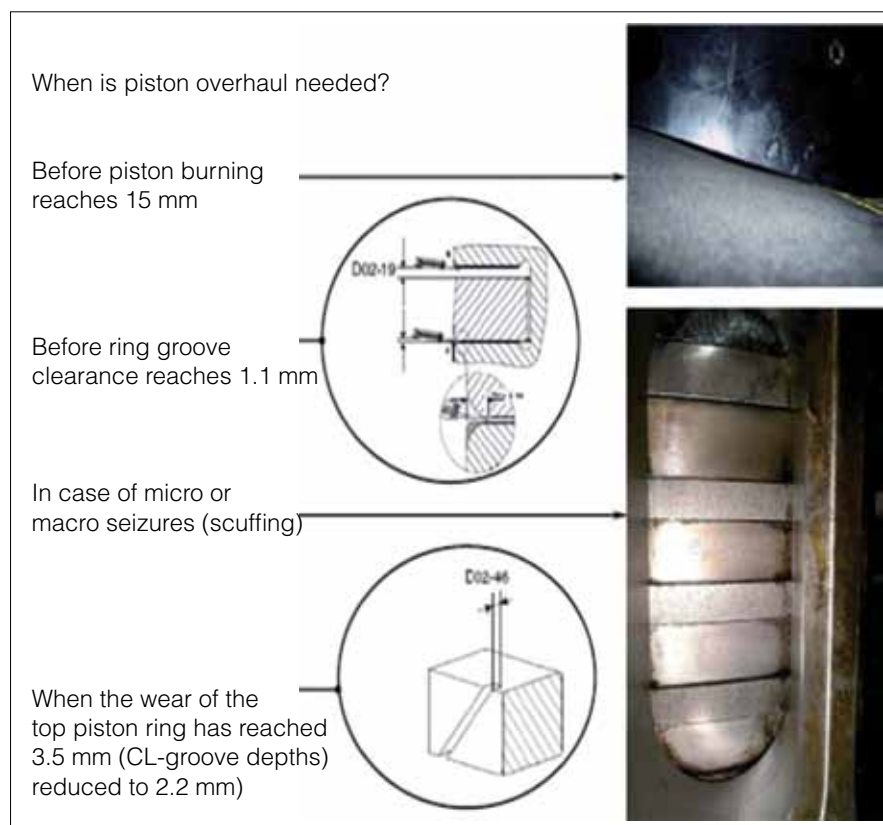


Fig. 41: K98 example. The four (4) important factors for piston overhaul

- Top piston ring wear, defined by measuring the remaining depth of the CL grooves.
- Ring groove clearances, measured with a feeler gauge.
- Estimated piston burnings on large bore engines, measured by means of a template via the scavenge ports.

Our standard sheets “Cylinder Condition Report” and “Inspection through Scavenge Ports” can be used, forming the ideal documentation for later review and for making trend curves for future wear forecasts, see Fig. 42.

The running surfaces of the piston rings are the best indicators of the cylinder condition in general. If the ring surfaces appear to be in good condition and free from scratches, micro or macro-seizures, the liner will also be in good condition.

If the liner appears damaged by active seizures (if the wave cut pattern has disappeared on the lower cylinder part visible through the ports), the rings will also be affected, and most likely the unit has to be overhauled.

As mentioned above, the wear on the top piston rings can be determined by measuring the remaining depth of the

CL grooves using a Vernier gauge, but wear can also be estimated visually simply by checking the size of the remaining rounding on the upper and lower edges of the running surfaces. From new, the rounding has a radius of 2 mm on 80/90/98-cm bore engines.

Thus, a simple visual inspection through the scavenge ports confirming that the rounding is still visible or partly visible is an indication that the wear limit has not been reached, and that many more hours are left before piston overhaul is necessary. For further information, we refer to our Service Letter SL07-483, Ref. [2].

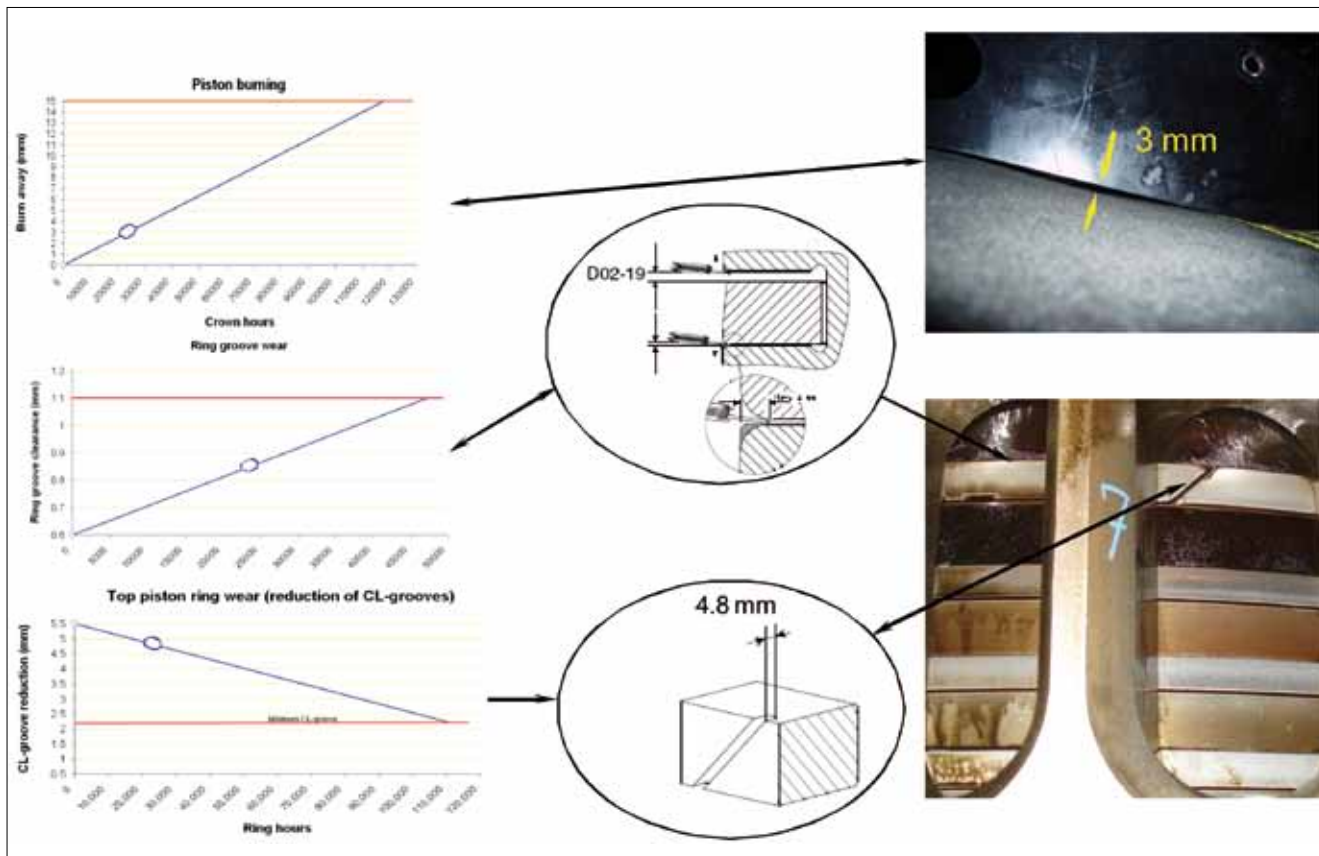


Fig. 42: 10K98MC-C, unit no. 7, at 23,500 hours without overhaul. The condition does not call for piston overhaul

CBO of Exhaust Valves

For the exhaust valve, the use of W-seat, see Fig. 43, and either Nimonic valve spindle or DuraSpindle has improved the overhaul intervals to run more than 32,000 hours. Fig. 44 shows examples of an excellent condition without overhaul with combinations of a W-seat/nimonic spindle and a W-seat/DuraSpindle achieved on an S60MC engine after 25,500 hours and 33,900 hours, respectively.

For the exhaust valve stem seal, the so-called Controlled Oil Level (COL) design, see Fig. 45, indicates that also stem seal overhaul intervals can be extended to 30,000-35,000 hours based on results from several test units on 98, 90 and 60-cm bore engines. This is illustrated by Fig. 46, showing an open-up inspection on a K98.

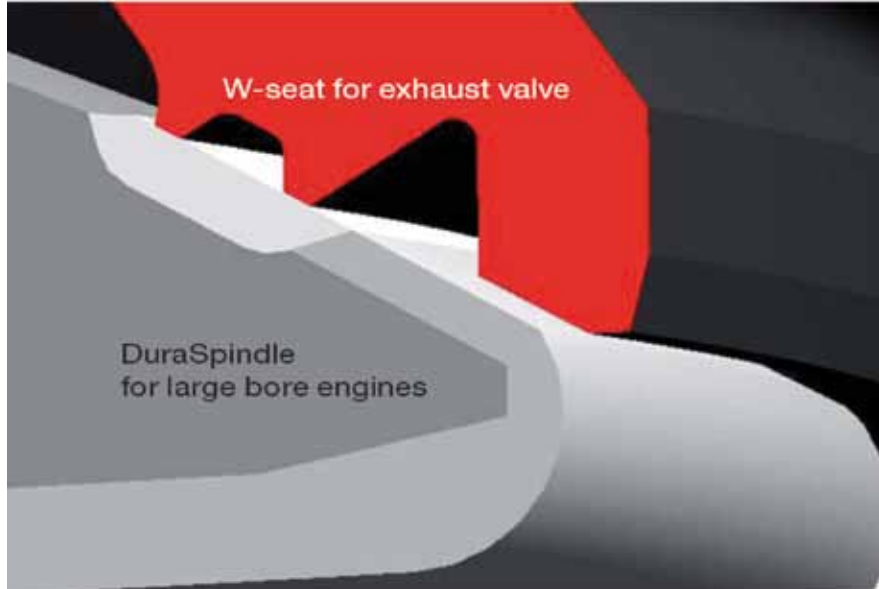


Fig. 43: W-seat and DuraSpindle combination



Fig. 44: W-seat in combination with Nimonic spindle and DuraSpindle on an S60MC

CBO of Bearings

Since the late 1990s, we have seen a positive development with respect to main bearing damage. Despite the large increase in the main bearing population on MC/MC-C engines, see Fig. 47, the reported number of damage incidents remains very low, see Fig. 48.

For other bearing types (crosshead and crankpin bearings), the damage frequency is also very low.

However, in a few cases, we have experienced severe damage, causing long-term offhire periods involving also costly repairs of the bedplate and/or the

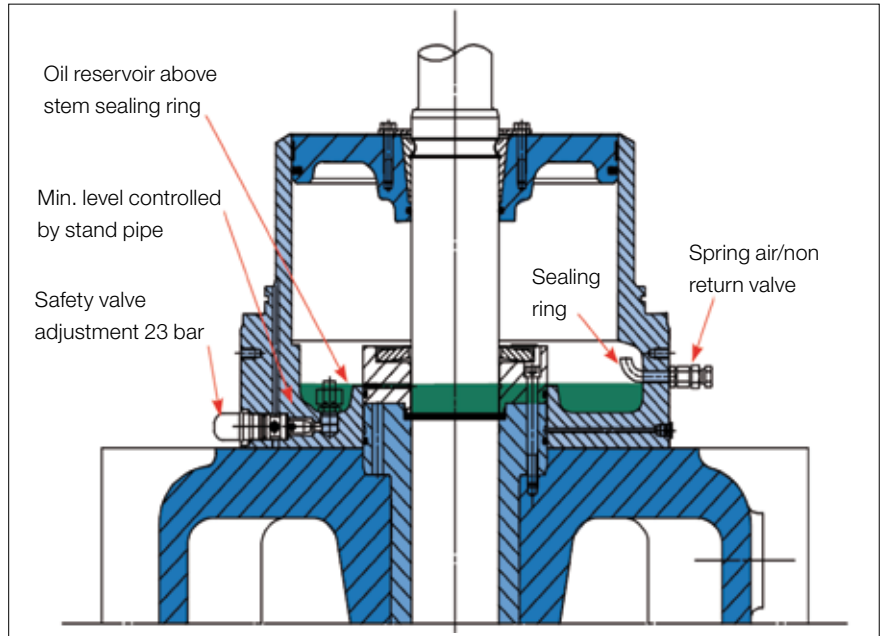
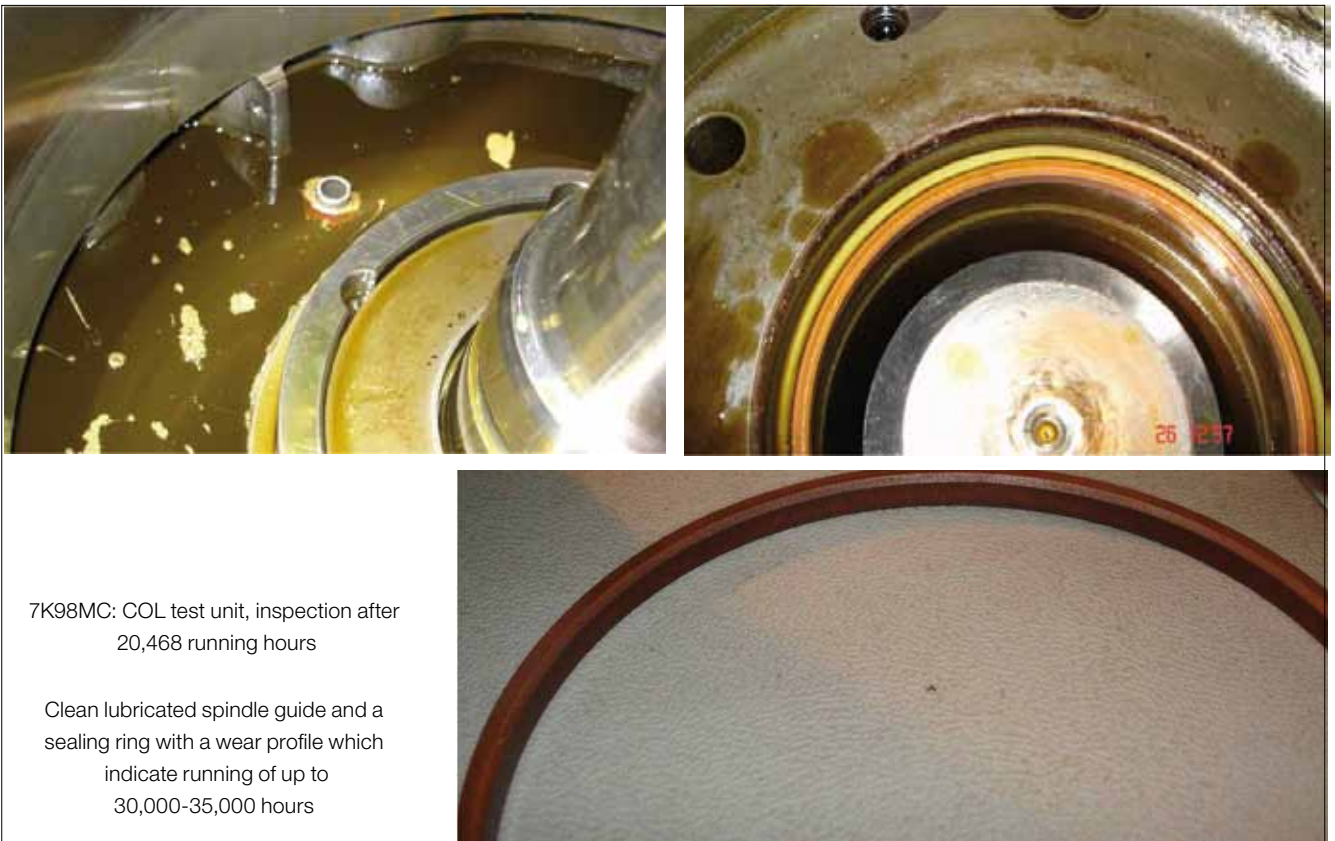


Fig. 45: Controlled Oil Level (COL) design



7K98MC: COL test unit, inspection after 20,468 running hours

Clean lubricated spindle guide and a sealing ring with a wear profile which indicate running of up to 30,000-35,000 hours

Fig. 46: Inspection of COL design on a K98MC

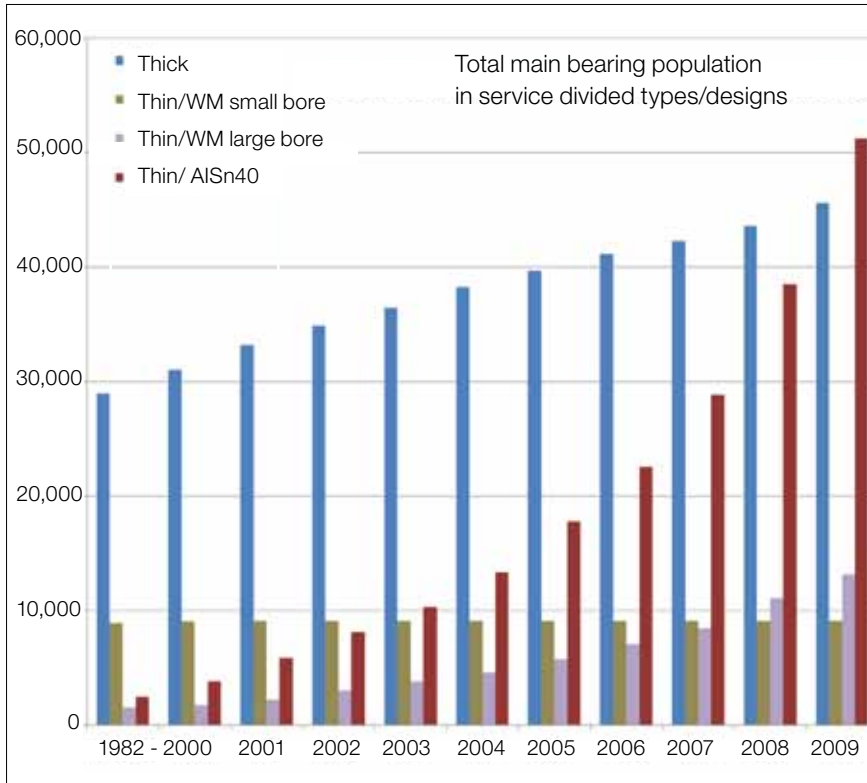


Fig. 47: Main bearing population 1982-2009 divided into bearing types

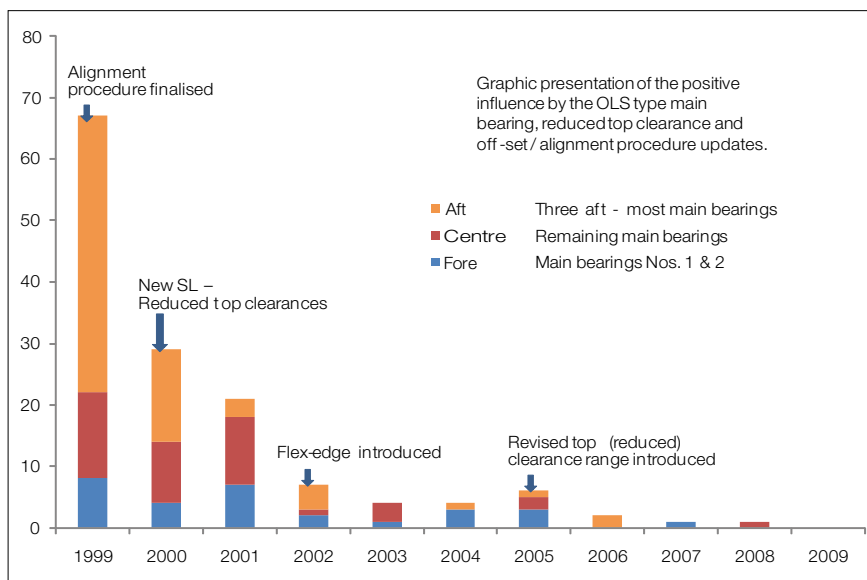


Fig. 48: Thick shell main bearing damage statistics

crankshaft. An example is shown in Fig. 49. In this case, the reason for the damage was incorrect assembly after an

open-up inspection of a main bearing after sea trial.

This sequence of events following open-up inspections of bearings is unfortunately being reported from time to time. We have therefore changed our instruction material, not calling for open-up inspection at regular intervals. In parallel, we have made so-called bearing wear monitoring (BWM) systems the standard on newly ordered engines. BWM systems can also be retrofitted on existing engines.

In principle, the BWM system monitors all major bearings (main, crankpin and crosshead) by measuring the distance to the bottom dead centre of the crosshead, see Fig. 50. The distance will decrease if wear occurs in one of the major bearings, and the BWM system can then give an alarm.

By monitoring wear in the major bearings, condition based monitoring (CBO) of bearings is introduced, and regular open-up inspections can be limited to fewer than previously. Optimally, open-up inspections should, if at all needed, only be carried out during dry-dockings or when indications (bearing metal in bedplate or BWM alarm) call for it.

This revised strategy will further limit the cases of severe bearing breakdowns.

Also water in oil (WIO) monitoring systems have been added to the standard instrumentation for newly ordered engines. This is especially important in relation to crosshead bearings with lead overlayer being sensitive towards corrosion due to a too high water content in the system oil.



6S70MC-C on maiden voyage

- Continued running for 1½ hrs after 1st alarm
- Main bearing incorrectly assembled after inspection
- 3½ month repair

Fig. 49: Main bearing damage on a 6S70MC-C

Time Between Overhaul (TBO) for Turbochargers

For turbochargers, all major makers are now promoting extended times between major overhauls, see Fig. 51. This means that for new turbochargers, it will be realistic to require major overhauls only during docking of the vessel. The overhaul intervals will then in many cases be five years.



Fig. 51: Modern turbocharger enabling more than 30,000 hours between major overhauls

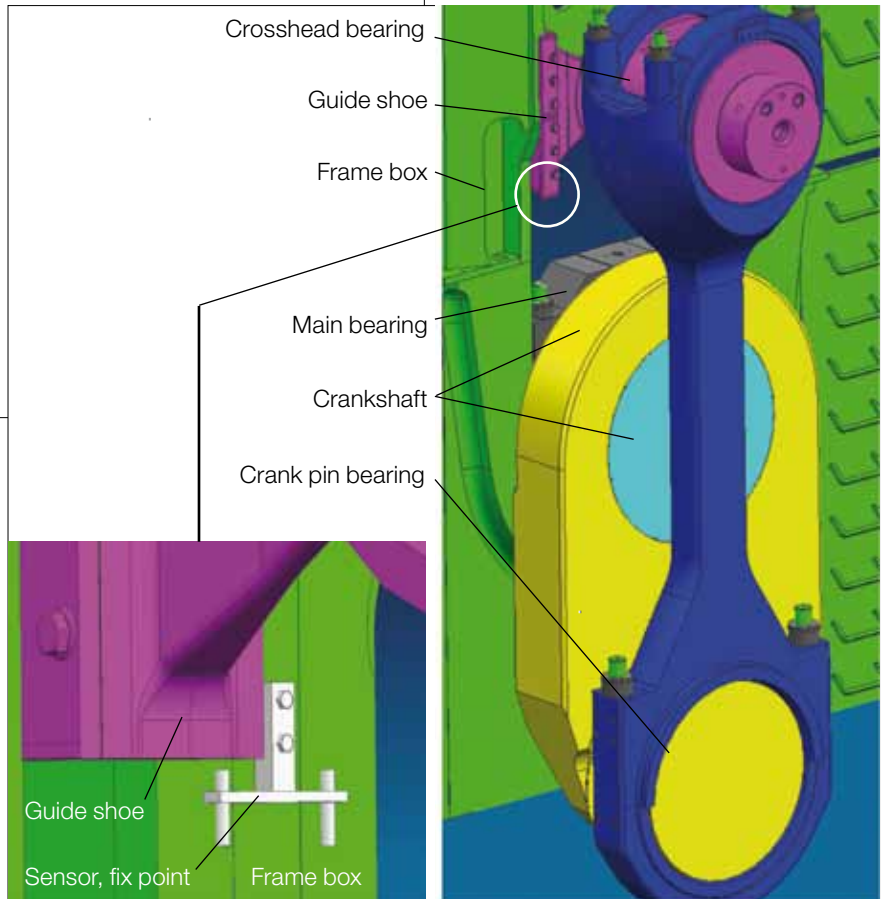


Fig. 50: Bearing Wear Monitoring (BWM), position of sensor

Low Sulphur Fuel Operation

MAN B&W low speed engines can operate on both heavy fuel oils (HFOs) with a varying amount of sulphur, marine diesel oil (MDO) and marine gas oil (MGO). All fuels are specified in accordance with ISO 8217 and CIMAC recommendation 21. Also biofuels (with separate fuel specification) are now used on MAN B&W low speed engines.

Local and international restrictions on sulphur emissions are the reason why an increased focus on low sulphur fuels is seen today. Sulphur emissions can be limited in two ways:

- By making rules for a maximum amount of sulphur in the fuel. Fig. 52 shows the “road map” for such legislation globally and locally in so-called SECAs (Sulphur Emission Control Areas)
- By applying abatement technologies on board the vessels allowing the vessels to continue operating on a high sulphur content HFO. The driving force for such technologies is the large price difference between various HFOs and distillates, see Fig. 53.

When running on low sulphur fuels, a number of issues of interest in relation to operational aspects can be mentioned. Many of these are dealt with in detail in our Service Letter, SL09-515/CXR, Ref. [3]. These issues are discussed one by one in the following.

Catfines in Low Sulphur HFOs

From a large number of bunker analyses, it can be seen that there is a tendency towards a higher amount of catfines in fuels with lower sulphur con-

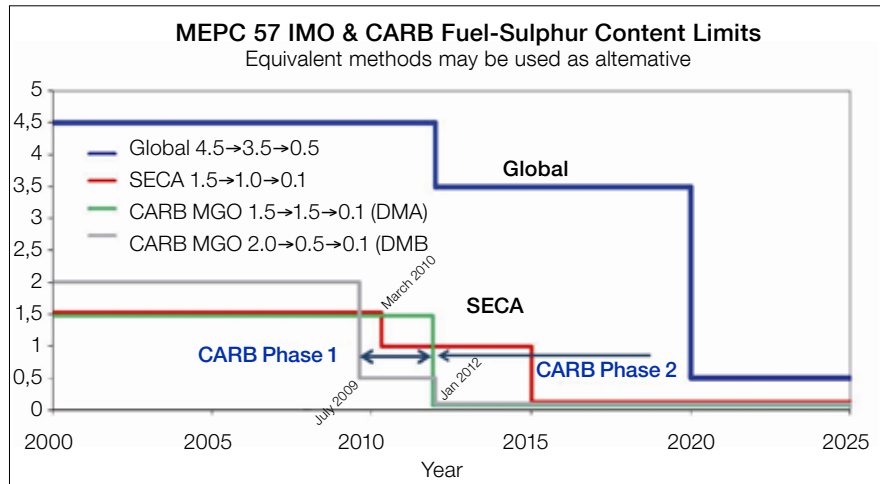


Fig. 52: Sulphur reduction ‘road map’

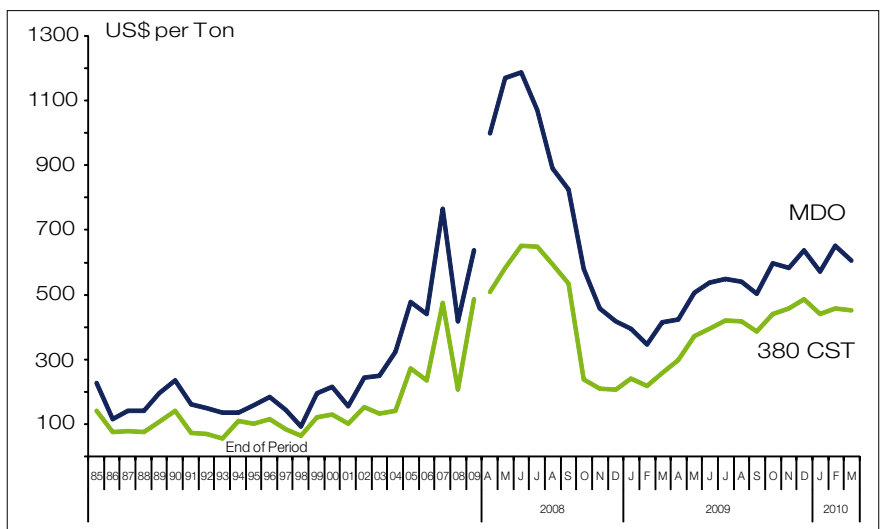


Fig. 53: Cost difference: HFO vs. distillates

tents. This requires increased focus on optimal function of the fuel treatment plants on board vessels operating on low sulphur fuels.

Cylinder Lubrication and Low Sulphur Fuels

It is well-established that MAN B&W low speed engines, to a certain degree, need cylinder oil feed rates proportional to the sulphur content in the fuel. This is due to the fact that we prefer to have a controlled amount of cold corrosion on

the cylinder liner wall. However, we also have other requirements for lubrication apart from controlling the acid neutralisation. These requirements presently put a minimum limit to the feed rate of 0.6 g/kWh. Fig. 54 illustrates the degree of over-additivation when operating on various cylinder oils (various BN numbers), and it can be seen that the need for lower BN cylinder oils will persist as fuel sulphur content limits are tightened.

Coping with Low Sulphur Fuel in the Design of the Combustion Chamber

Design-wise, we can lower the cylinder liner temperature by increasing the cooling intensity of the cylinder liner. By doing this, we can provoke an increased amount of cold corrosion when operating on low sulphur fuels. For some engine types, we have introduced such colder cylinder liners. Also the piston ring pack has undergone changes, which will optimise the performance in a low sulphur fuel regime. We have introduced cermet coating on piston rings Nos. 1 and 4 to make them less dependent on cold corrosion on the cylinder liner wall.

Viscosity Issues

Viscosity has been dealt with in detail in the above-mentioned service letter on distillate fuel operation, Ref. [3]. When using distillates in order to adhere to the rules for sulphur emission, viscosity often becomes an issue. Our updated instructions regarding fuel viscosity are illustrated in Fig. 55. Detailed recommendations regarding checks to be made before entering ports and other narrow water passages are outlined in the service letter, Ref. [3]. Also fuel cooling systems are presented. The aim of these systems is to lower the temperature in order to maintain viscosity of 2 cSt at engine inlet.

Burning Characteristics for Low Sulphur Fuels

From time to time, we hear that slow burning characteristics of fuels give rise to concern. These slow burning cases are more frequently seen in cases of low sulphur fuels. However, such slow burning characteristics will not affect MAN B&W low speed engines. The reason is

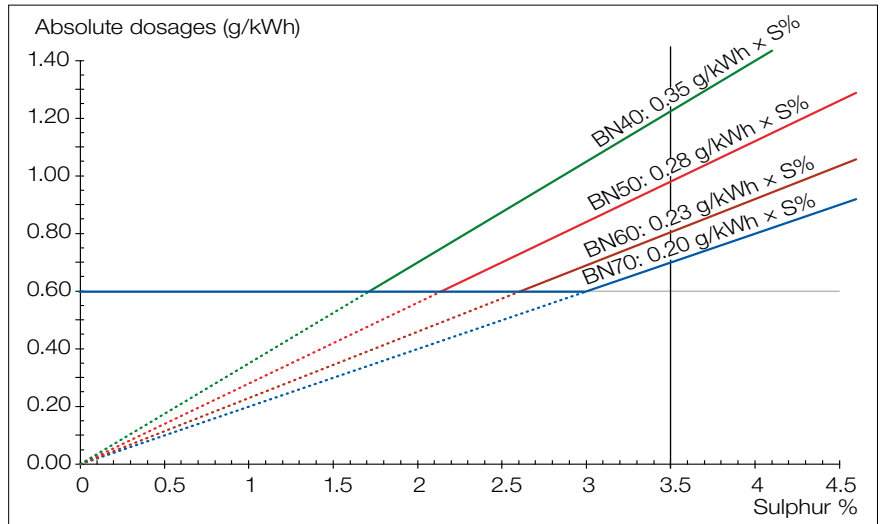


Fig. 54: Low sulphur fuel operation: need for lower BN oil in cylinder

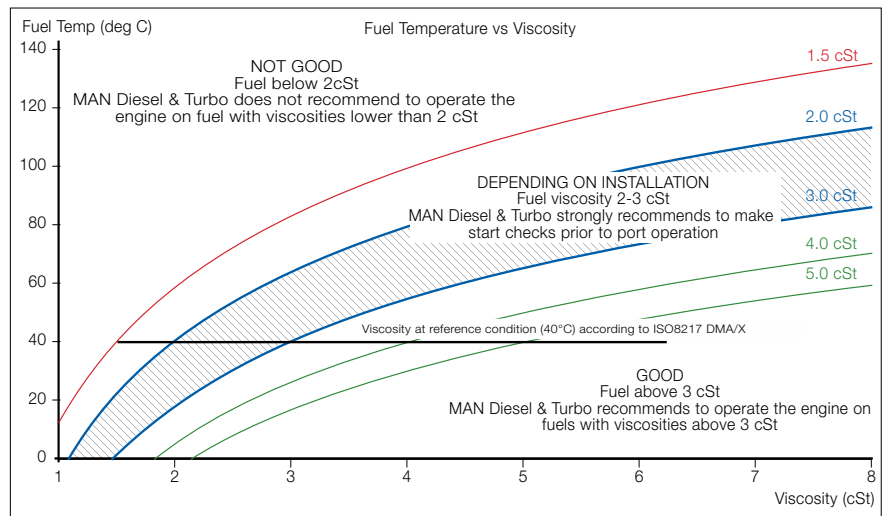


Fig. 55: Fuel temperature vs. viscosity

the slow speed concept with the relatively long combustion duration. However, for four-stroke medium and high speed engines, the matter is different and slow burning fuels may cause trouble on these engine types.

Abatement Technologies

We are involved in various projects where scrubbers are used to clean the exhaust gas for sulphur and other particles. The driving force behind these

projects is the wish to maintain operation on the cheaper HFO, see Fig. 53.

Gas Burning Low Speed Engines: ME-GI

Another future way of avoiding sulphur in the exhaust is to change from diesel fuel to natural gas. In this way, SO_x can be reduced by typically 90%. We believe that natural gas will be the future fuel not only on LNG carriers, but probably also on other commercial types of vessels.

Laying-Up of Vessels

Because of the current economic condition in the world, we have been requested to renew instructions in relation to laying-up of vessels. We have recently issued two Service Letters on this topic. The first one, SL09-502/SBJ, Ref. [4], deals with the so-called hot laying-up where auxiliary engines are kept in operation continuously in order to generate the necessary power to run, from time to time, as for example the main lube oil pumps for the main engine. The second Service Letter, SL09-510/SBJ, Ref. [5], deals with cold laying-up of vessels where also the auxiliary engines are closed down. In this case, power for dehumidifiers, various pumps and the turning gear for the main engine is typically taken from an on deck containerised power pack.

A number of items are to be considered in relation to the method of laying-up and some decisions are needed. Some of these items are:

- Mode of laying-up (hot or cold)
- Maintenance work to be done during laying-up period
- Level of manning during laying-up period
- Customised laying-up check list (to assist when ending laying-up period)
- Estimate of time to re-establish engine operation after laying-up period.

In detail, our service letters deal with:

A. Corrosion and how to avoid it. The use of dehumidifiers is discussed.

The use of preservation oils on machined surfaces inside as well as outside the engine compartments is described. During the monthly turning of the main engine, preservation oils have to be re-established. Both main engine and auxiliary engines are dealt with in this respect.

B. Detailed instructions for preservation of main engine and auxiliary engines, including preservation of supply systems, are outlined in the service letters. Special attention to bacterial growth in fuel oils may be needed with use of biocides to control the bacteria level.

C. Turbocharger preservation is also dealt with. Here, makers distinguish between short laying-up periods, below 6 months, where the rotor may stay in the T/C casing, and laying-up periods longer than 6 months where the complete rotor has to be dismantled, cleaned and stored according to the maker's instructions.

D. Conservation of electrical components has its own chapter in the service letters. This is especially relevant when dealing with laying-up of electronically controlled ME engines.

E. Recommended routine checks during the laying-up period is described in detail. Examples of checks are daily recording of the humidity level, monthly oil and cooling water circulation, pre-lubrication of intermediate and propeller shaft bearing before turning of the main engine, monthly check and recording of water content in the lube oil and monthly check of the cooling water inhibitor level.

We are still gaining relevant experience in relation to laying-up of vessels and a revision/extension of the service letters is expected to be launched during 2010.

Slow Steaming

Slow steaming has also become very relevant due to the financial situation in the world. Also on this topic, we have issued service letters relevant for MAN B&W low speed engines, SL08-501/SBE, Ref. [6], and SL09-511/MTS, Ref. [7]. The reason for the desire for slow steaming can be seen in Fig. 56.

A vast amount of fuel can be saved per tonnes-mile of seaborne travel when reducing the ship speed and thereby the engine load.

Fortunately, we have been able to support continuous operation down to 10% load without any engine modifications, except the use of slide fuel valves.

There are various means to optimise operation at low load. Some of these are:

- Turbocharger with VTA, Fig. 57
- Part load optimisation, Fig. 58
- Sequential turbocharging, Fig. 59
- Turbocharger cut-out, Fig. 60
- Exhaust gas bypass

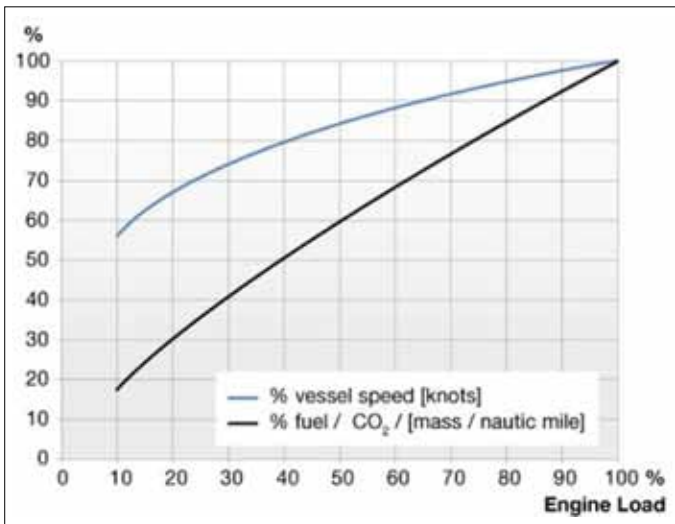


Fig. 56: Low load operation

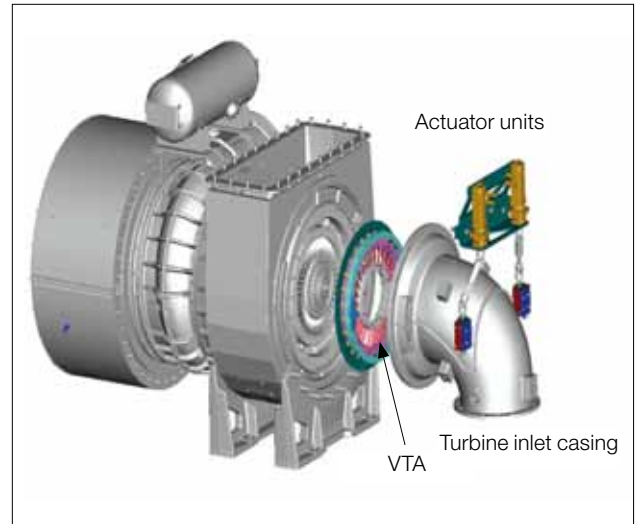


Fig. 57: Turbocharger with variable turbine area (VTA)

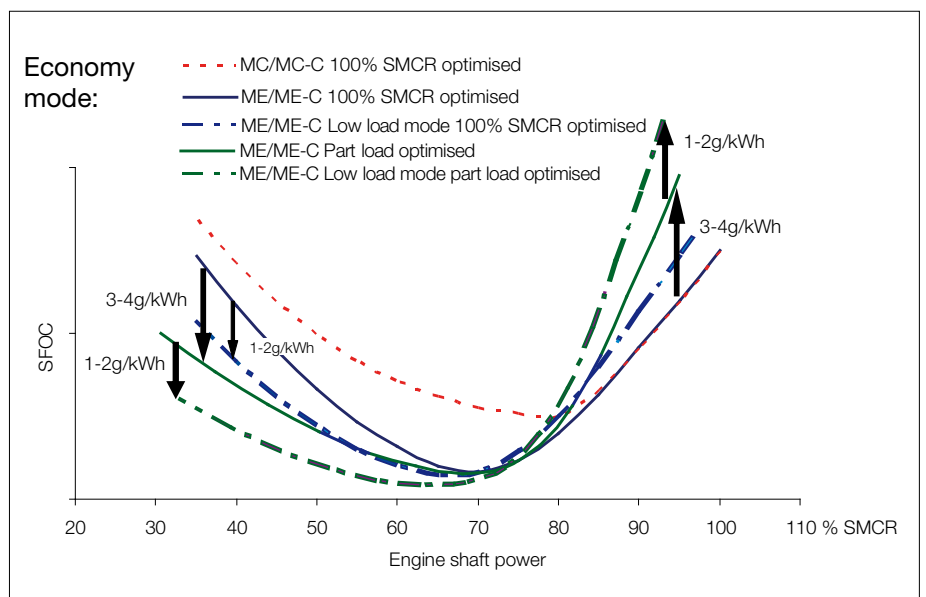


Fig. 58: Part optimisation ME/ME-C and MC/MC-C

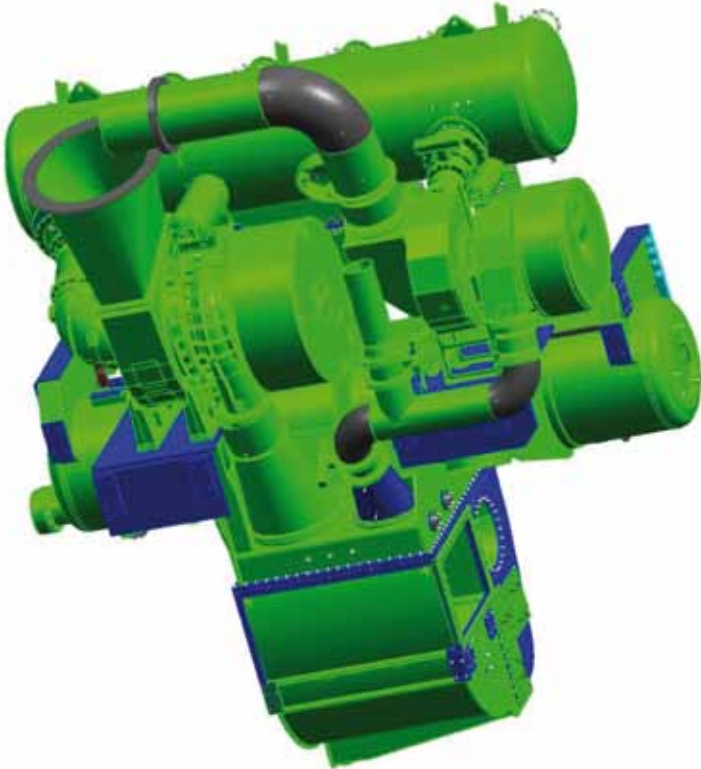


Fig. 59: Sequential turbocharging

VTA Service Experience

The first VTA turbocharger has been in service for well over 10,000 hours on a 6S46MC-C engine on a vessel operating on heavy fuel. Service experience has been very good and the test continues. More details with respect to service experience can be found in the Worldwide Turbocharger Conference 2009 paper, Ref. [8].

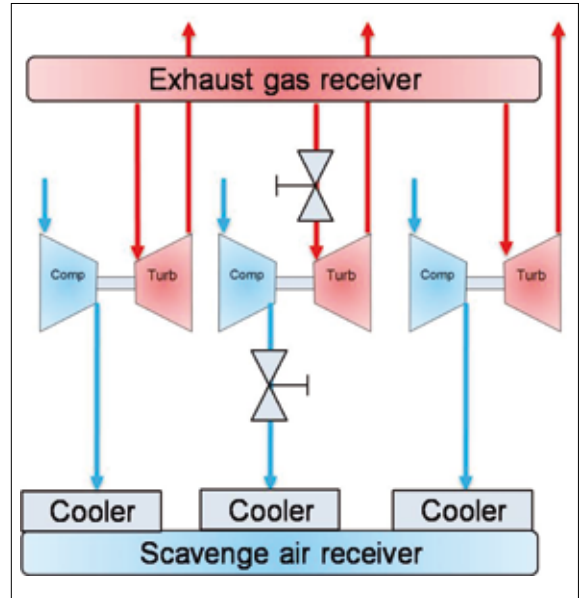


Fig. 60: Turbocharger cut-out

T/C Cut-out and Swing Gate Valve Experience

A flexible T/C cut-out system assisted by an arrangement of so-called swing-gate valves, Fig. 61, has been successfully tested on a 12K98ME engine with three turbochargers. The middle turbocharger was retrofitted with swing-gate valves.

In late December 2009, the engine and especially the turbocharger subjected to T/C cut-out were inspected. At the time of the inspection, the T/C cut-out system had been in operation for a total of 1,637 hours, and cut-out mode had been realised for 1,270 of these hours. During the T/C cut-out operation, the turbocharger was at standstill, lubricated and with sealing air applied. The axial and the radial bearings in the

turbocharger were both in normal, good condition.

The service test described above indicates that a flexible T/C cut-out system is an option when optimising for low-load operation and that with a limited number of countermeasures it is possible to leave the T/C rotor at standstill in the turbocharger.

All methods will increase the scavenge air pressure at part load. The methods are more or less easy/costly to retrofit on vessels in service. However, it should be kept in mind that this optimisation only gives the last marginal benefit of slow steaming. The major benefit comes just from moving the “speed handle” down.

Conclusion

This paper has given an update on recent service experience and, in addition, it has touched upon other areas of interest for the operators of MAN B&W low speed engines forced upon the industry owing to legislation as well as being a result of the prevailing financial situation in the world today. We will continue our efforts to adopt and optimise our product under these circumstances.

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- Ref. [2]: SL07-483/HRR, Condition-based Piston Overhaul, August 2007
- Ref. [3]: SL09-515/CXR, Guidelines on Operation on Distillate Fuels, September 2009
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- Ref. [6]: SL08-501/SBE, Low Load Update – Down to 40% Load, October 2008
- Ref. [7]: SL09-511/MTS, Low Load Operation 10% to 40% Engine Load, May 2009
- Ref. [8]: “MAN Turbocharger Applications with Variable Turbine”, by Dr. Herbert Schmuttermair, Senior Manager, Turbocharger Application, MAN Diesel & Turbo, Worldwide Turbocharger Conference, Hamburg, 2009

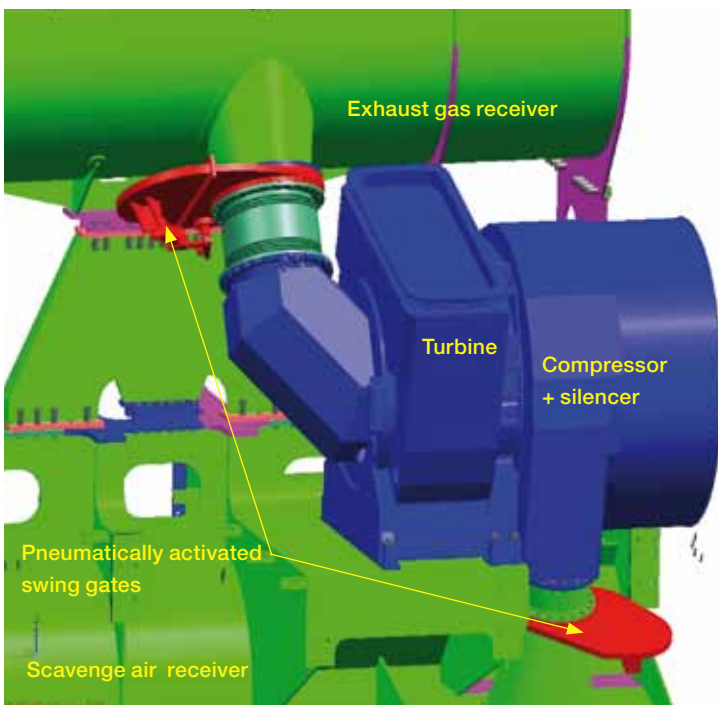


Fig. 61: Pneumatically activated swing gates

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